Energy Research and Development Division

FINAL PROJECT REPORT

Building Energy Efficient Cooling and Heating

California Energy Commission

Edmund G. Brown Jr., Governor

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PREFACE

The California Energy Commission's Energy Research and Development Division supports energy research and development programs to spur innovation in energy efficiency, renewable energy and advanced clean generation, energy-related environmental protection, energy transmission and distribution and transportation.

In 2012, the Electric Program Investment Charge (EPIC) was established by the California Public Utilities Commission to fund public investments in research to create and advance new energy solutions, foster regional innovation and bring ideas from the lab to the marketplace. The California Energy Commission and the state's three largest investor-owned utilities—Pacific Gas and Electric Company, San Diego Gas & Electric Company, and Southern California Edison Company—were selected to administer the EPIC funds and advance novel technologies, tools, and strategies that provide benefits to their electric ratepayers.

The Energy Commission is committed to ensuring public participation in its research and development programs that promote greater reliability, lower costs, and increase safety for the California electric ratepayer and include:

- Providing societal benefits.
- Reducing greenhouse gas emission in the electricity sector at the lowest possible cost.
- Supporting California's loading order to meet energy needs first with energy efficiency and demand response, next with renewable energy (distributed generation and utility scale), and finally with clean, conventional electricity supply.
- Supporting low-emission vehicles and transportation.
- Providing economic development.
- Using ratepayer funds efficiently.

Building Energy Efficient Cooling and Heating is the final report for the Building Energy Efficient Cooling and Heating project (PIR-12-029) conducted by Altex Technologies Corporation. The information from this project contributes to the Energy Research and Development Division's EPIC Program.

For more information about the Energy Research and Development Division, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-327-1551.

ABSTRACT

Current systems that generate electricity from low-grade heat are not very efficient. The Building Energy Efficient Cooling and Heating project set out to combine an organic Rankine power cycle with a refrigeration cycle. It would convert waste heat or solar thermal energy directly into heating and refrigeration outputs, thereby eliminating energy losses due to conversion to, and reconversion from, electric power. The design used relatively low-cost components commonly used in the air conditioning and refrigeration industry. An important component of the project was a novel scroll-based integrated expander/compressor device. Researchers designed and built a full-scale system with expected output of 60,000 British thermal units of cooling per hour, and 190,000 British thermal units per hour of water heating. However, the expander/compressor could not be made to reliably start and run, which prevented completion of the planned steady state testing. Economic analysis of the system using engineering assumptions supported a 4-year payback when driven by waste heat, and a 13-year payback when driven by solar thermal energy. If it worked as designers projected, engineering calculations indicate that at full capacity the system could reduce greenhouse gas emissions by 722 pounds per day and save up to $17,353 per year, with most of those benefits provided by reduced natural gas consumption for water heating.

Keywords: building energy efficiency; refrigeration; cooling; combined cooling, heating, and power; advanced heat exchangers; mechanical vapor compression; refrigeration; water heating; scroll expander.

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Introduction

Technologies exist to harvest waste heat from commercial and industrial equipment such as boilers and convert that heat to electrical power. However, these technologies often have high initial costs and low outputs, which cause long payback times. The electrical output of these systems offsets grid power consumption, but the buildings where they are installed often use grid power to drive air conditioning or chiller equipment. Energy conversion losses from the electric generator, and from electric motors driving the refrigeration compressors, reduce the overall efficiency of this process. The Building Energy Efficient Cooling and Heating (BEECH) technology was conceived to eliminate these conversion inefficiencies by generating cooling directly, without the intermediate conversion to electrical power. The BEECH concept would use a heat engine using an organic Rankine cycle (a closed-cycle system where a working fluid circulates through an evaporator, turbine, condenser, and a pump to convert heat into work), employing organic refrigerants (those that contain carbon) at relatively low temperatures. The heat engine would be directly coupled to a refrigerant compressor used in a refrigeration cycle to produce cooling. The system, as conceived, could also supply condenser water hot enough for space heating or domestic hot water. The concept was to use inexpensive, commercially available components to keep costs down, and to design the system so as to be commercially applicable to a number of common waste heat sources or solar-heated water.

Project Purpose

The project team sought to design, construct, and demonstrate a Rankine-cycle-based machine that could use heat normally wasted in industrial and commercial facilities, or heat from solar or geothermal sources, to produce usable cooling and heating for space conditioning, refrigeration, or other purposes, at an attractive cost.

Project Process

The team members started by developing a simulation model of their heat engine. Then they identified suitable building types based on heating and cooling needs and available heat sources, including seasonal and diurnal availability of heat to fuel their engine. Once a specific use case was identified, they set about designing a practical engine for that use case, using reliable but relatively inexpensive, commercially available components where possible.

The team constructed a prototype and a testing setup for the proof-of-concept, and tested performance of the various sub-systems. Unfortunately, they were not able to get their prototype expander/compressor to start and run reliably, even after repeated efforts with various modifications. They ultimately completed their analysis of the concept based on projected performance from engineering calculations, and wrote the final report.

Project Results

Analysis showed that a 60,000 British-thermal-unit (Btu)-per-hour cooling/190,000 Btu-per-hour hot water system was best matched for the commercial building use-case that was selected. Testing of a partial, subscale solar thermal system showed the capability of the selected components to vaporize the compressed working fluid at the appropriate temperature. Additional subcomponent tests identified pumps, flow meters, and refrigeration system components that could be used in a full-scale solar- or waste heat-driven system. The project team designed and fabricated a full-scale waste heat-driven system that included an integrated expander/compressor based on the same scrolls used in refrigeration scroll compressors. Unfortunately, while these devices could be mass-produced at low cost, the expander/compressor could not be made to start and run reliably within the scope of this project.

An economic analysis of the system, supported by analytic models, industry sources, and limited experimental results, showed that if a BEECH system could be made to function with a 60,000 Btu-per-hour cooling and 190,000 Btu-per-hour heating capacity, it would have a 4-year payback when driven by waste heat, and a 13 year payback when driven by solar thermal energy.

Project Benefits

When installed on thermal equipment with greater than 75 percent thermal efficiency, preliminary calculations show that BEECH, if functional, could increase thermal efficiency by 10 percent, and reduce greenhouse gas emissions by 722 pounds per day. After the initial four-year payback period, the system would provide an operating cost reduction to the building in which it is installed. Using the waste heat system as an example, researchers anticipate a $17,353-per-year benefit, with most of the benefits accruing from avoiding the use of natural gas to heat water, which would instead be provided by the hot water output of the system.

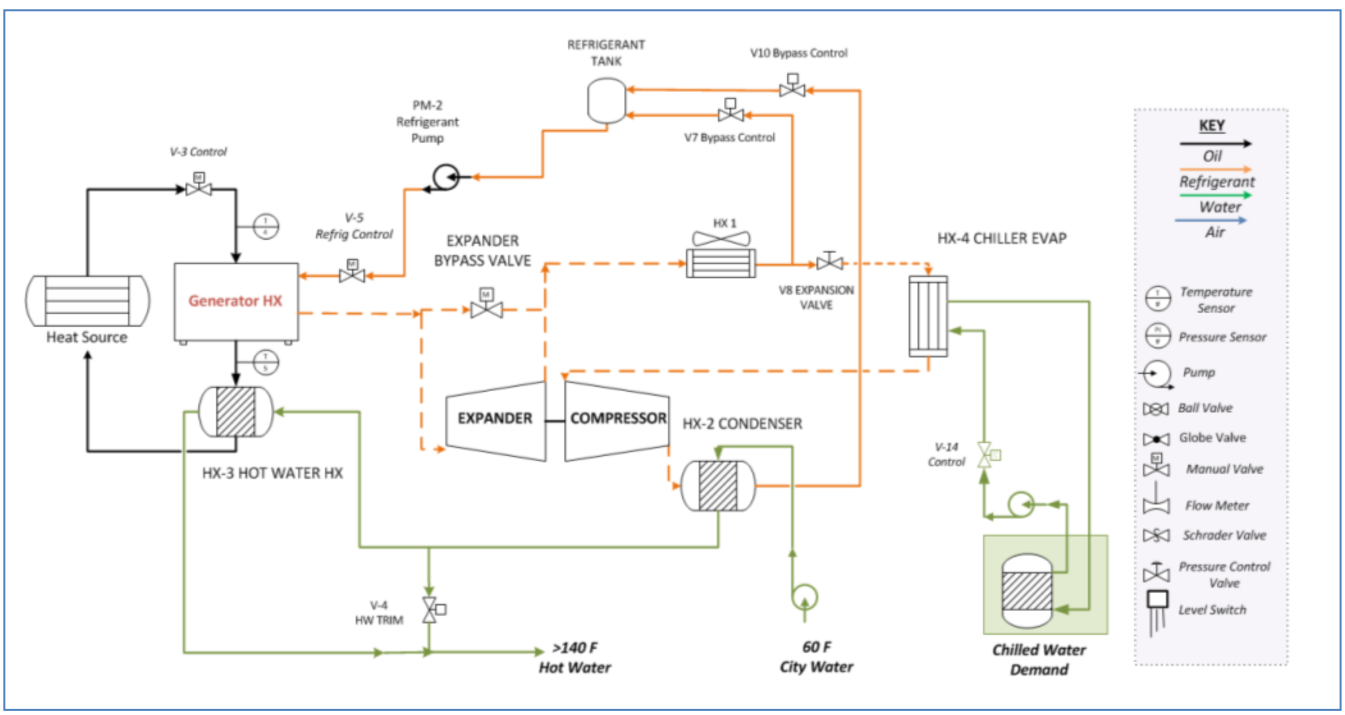
The project development work in advanced heat exchangers and the novel expander-compressor have the potential to support advanced energy efficiency and waste heat recovery systems in California and beyond. As Altex and its California-based partners continue to develop these technologies there is a potential for additional manufacturing jobs to be created in California, and for these systems to be installed and supported in the California market, providing a benefit to ratepayers of investor-owned utilities.

CHAPTER 1: Project Overview

The Building Energy Efficient Cooling and Heating (BEECH) technology uses either waste heat or solar thermal energy to generate cooling and heating for commercial buildings. The technology is applicable to industrial and large residential sites. Other waste heat and solar thermal-utilization technologies that produce power can be installed in large commercial buildings, but they often have high initial costs, which cause long payback times. The electrical output of these systems does offset grid power consumption, but the facilities in which they are installed also use grid power to drive air conditioning or chiller equipment. This results in double efficiency losses—once in converting and conditioning power from the generator device, and then again by converting electrical power to shaft power to drive the cooling equipment. BEECH seeks to eliminate that double inefficiency by generating cooling within the system, without the intermediate conversion to electrical power. To achieve quick payback times, BEECH was designed to use lower-cost system components than those competing power systems.

Figure 1 shows the final process design that the Altex team designed and refined in this project.

Figure 1: Final Process Design



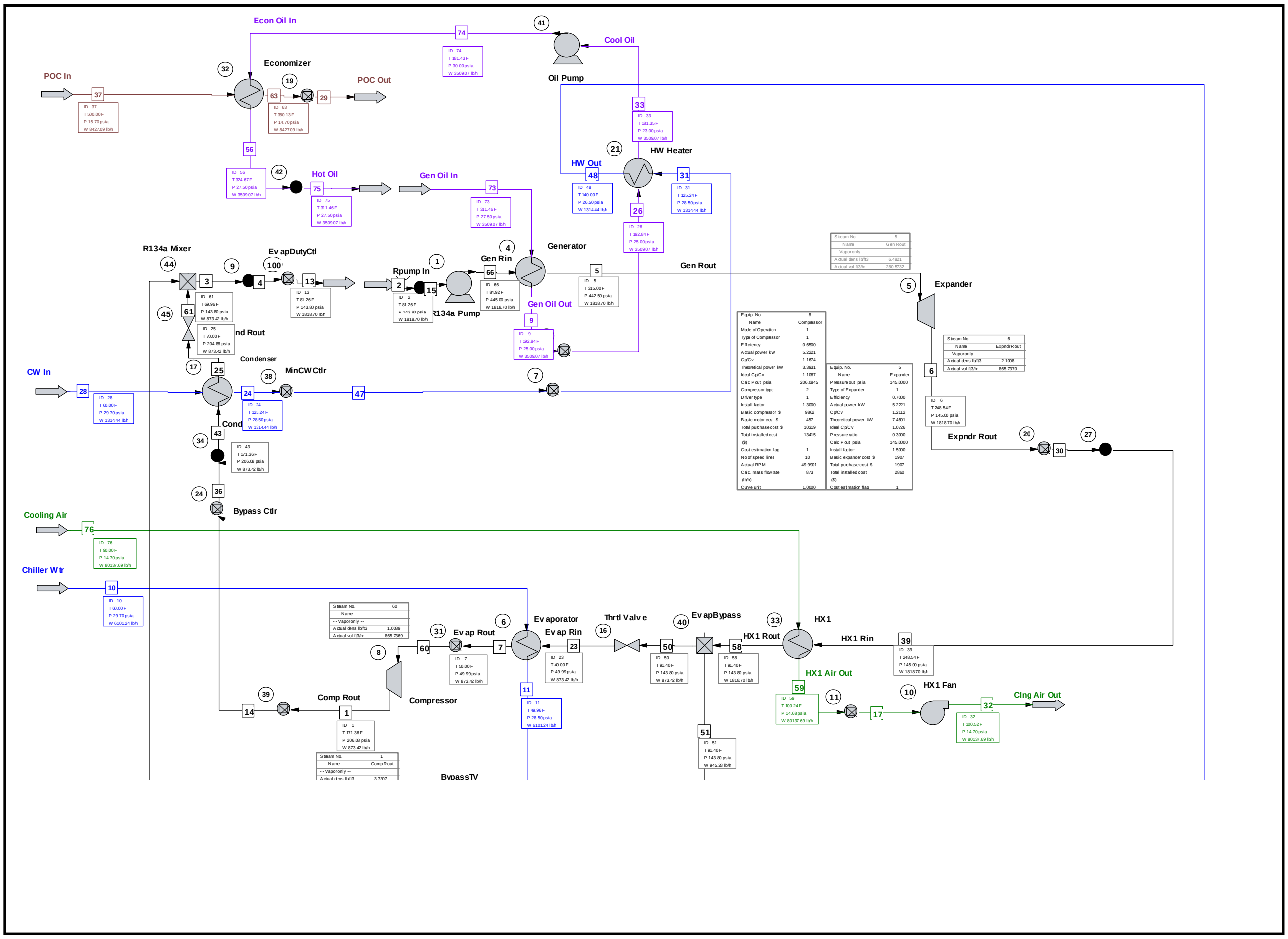
Source: Altex Technologies Corp.

The energy (solar or waste heat) is transferred from the heat source to a high pressure liquid in the generator heat exchanger (HX). The liquid boils and becomes a high pressure, superheated vapor. The vapor is expanded in an expander, which is directly coupled to a compressor. The expanded vapor is then condensed in HX1. A fraction of the condensed vapor is returned to a refrigerant tank or reservoir, and the remainder is delivered to a thermal expansion valve. From there, the BEECH system functions much like a conventional mechanical vapor compression cycle, with flow through an evaporator, compressor, and condenser. The fluid is then returned to the reservoir, from where it is pressurized and pumped to the generator.

The process is essentially a Rankine power cycle mated to a refrigeration cycle, but there is a key difference between this and other variants proposed or tested by other researchers. The cycle operates with a single working fluid, but mass flow is not constant throughout the system. The bypass flow, which is the fraction of condensed liquid after the expander that is not sent to the refrigeration cycle, can be varied to control the speed of an integrated expander/compressor. For the paired unit to operate at a stable speed, the work output of the expander must match the work consumption of the compressor. Otherwise, the unit will speed up or slow down. Since environmental and site demands will vary (for example, the ambient air temperature will affect power cycle condenser temperature), the bypass flow can be adjusted to balance the work of the two units. The overall hot water or cooling output of the system can then be varied by changing: the oil or glycol flow from the heat recovery or solar thermal collector; the water and air flow rate to the condensers; or the speed of the expander/compressor. Operation at the designed speed (3000-3600 rpm) and pressure ratios will likely produce the most efficient conversion of heat to cooling under most conditions. Altex performed chemical process modeling of the BEECH system using the commercial CHEMCAD process modeling tool to arrive at the final design. For clarity, an explanation of the process is included here, and matches the illustration in Figure 2.

The waste heat variant of BEECH uses a heat recovery heat exchanger (HRHX, also known as an “economizer” in the boiler industry) with a finned coil to transfer heat from the exhaust of natural-gas fired devices, such as boilers and water heaters, to a low-vapor-pressure thermal oil. The solar thermal variant of BEECH uses evacuated-tube solar collectors to transfer heat into a glycol/water solution.

Figure 2: BEECH Process Design—CHEMCAD Flow Sheet



Source: Altex Technologies Corp.

CHAPTER 2:
Site Specification

To determine the proper system sizing for both the cooling and heating outputs, Altex engineers analyzed waste heat- and solar-heat driven BEECH systems for commercial building applications of high interest for energy efficient enhancements. The proposed system capacity for a "typical" commercial building was 15 tons cooling (180,000 Btu/hr) and 3.6 therms/hr heating (360,000 Btu/hr), and so the site specification activities also reviewed this assumption, to ensure that the BEECH system that would be designed, built, and tested under this project would be of maximum benefit to commercial facilities of interest.

Simultaneous Cooling and Heating Demand

In support of the Site Specification report, the team downloaded detailed heating and cooling data from the *California Commercial End Use Survey* (CEUS) and screened that data for the broad building categories that offered good BEECH installation potential. That data was best suited for evaluation of the waste heat recovery variant of BEECH, since the thermal input to the buildings could be acquired from the same data set as the cooling. Solar BEECH's potential thermal input will be driven by two site-specific factors: solar insolation and the available installation space for the thermal collectors, as discussed below.

Figure 3 displays sample data, as graphed by the CEUS website. January and July data sets were both reviewed, to evaluate the extremes of heating and cooling demand. The data reflects an entire building segment in a given Investor Owned Utility service area, and could be evaluated by the following criteria for screening purposes:

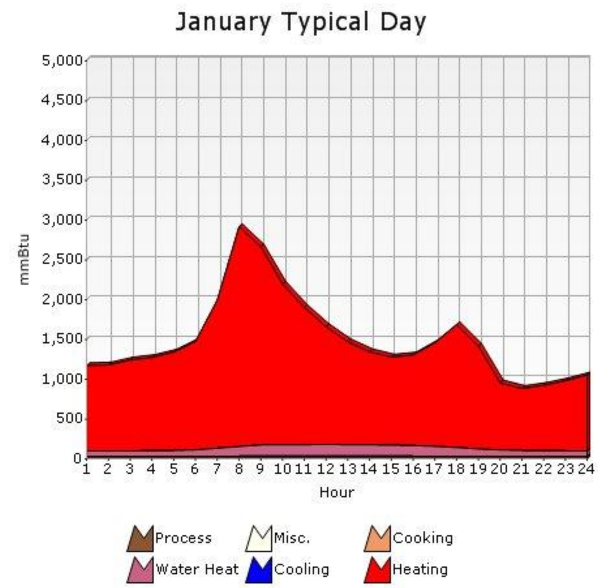
- Simultaneous, day-long demand for BEECH's useful outputs: cooling and hot water
- Day-long thermal input for space, process, and water heating, indicating continuous waste heat production
- Adequate magnitude of cooling and hot water demand to allow BEECH to operate as a base-load device

The latter criteria proved difficult to evaluate numerically, since the data represented an entire building classification.

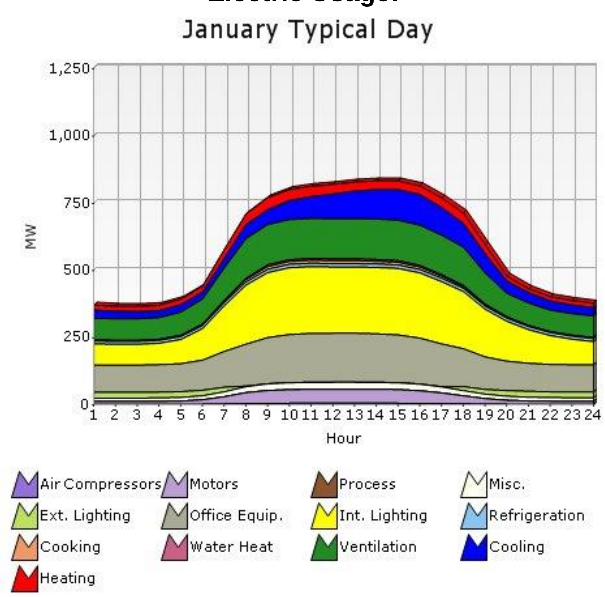
The numerical data underlying graphs like those shown in Figure 3 were downloaded for various business segments, and the project goals for cooling and hot water were used to evaluate potential useful outputs, based on the project's cooling and heating goals. The results are shown in California End Use Survey Figure 4 through Figure 7. Since the data reflects a complete market sector, the magnitude of the BEECH predictions reflects the total available market. The more useful analysis result from these graphs is the relative demand and production of the various energy uses, as indicators of the proportions that might be present in individual buildings within that sector.

Figure 3: California Commercial End Use Data—Sample Inputs Used for Screening

Gas Use:

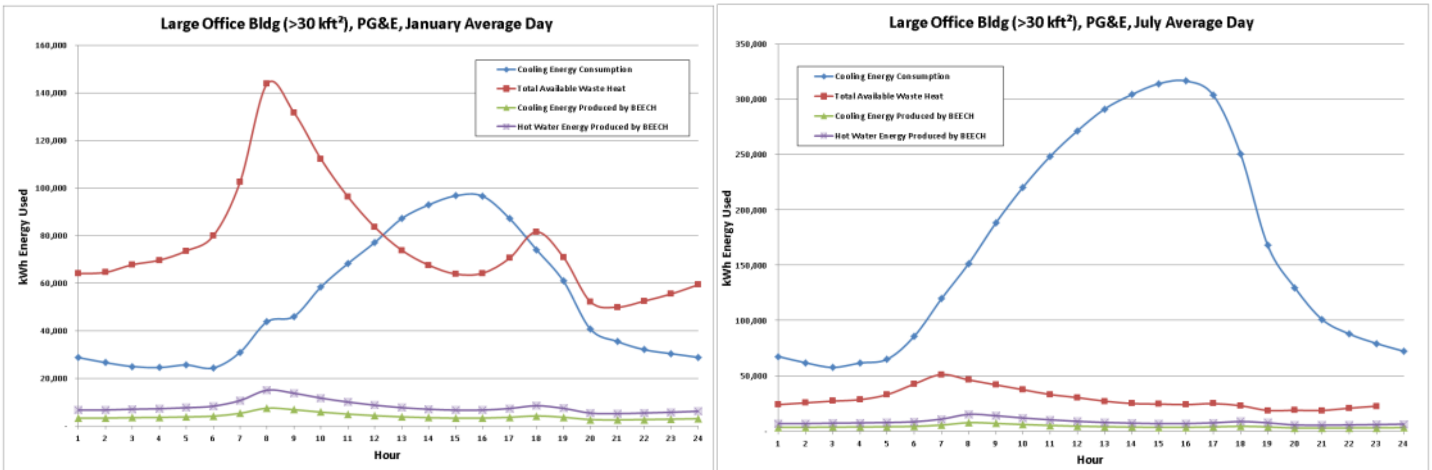


Electric Usage:



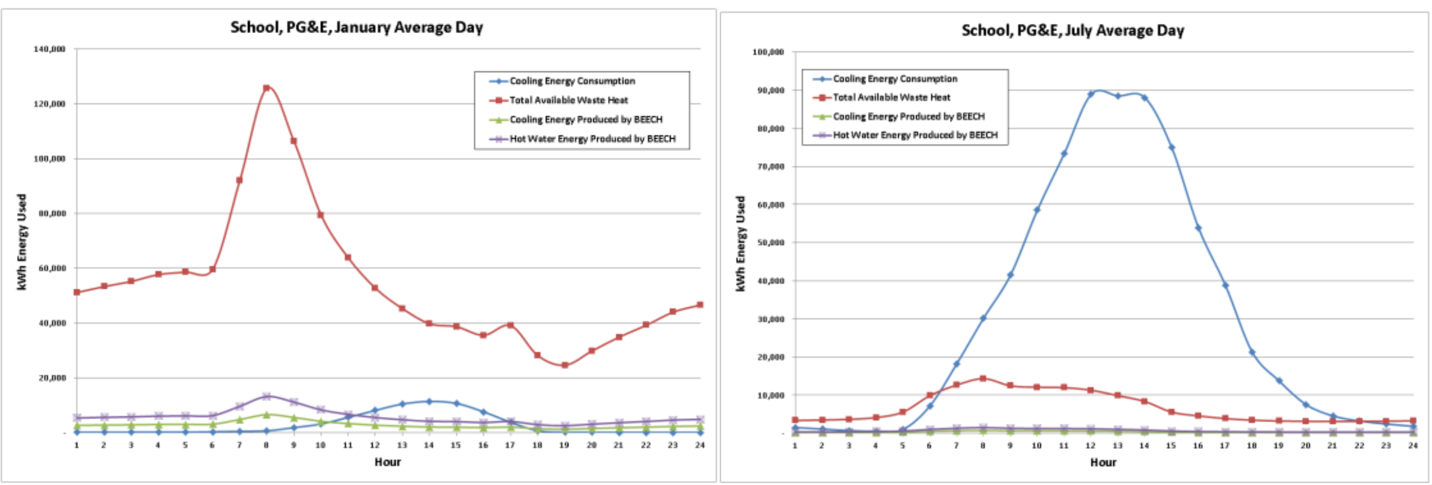
Source: California End Use Survey

Figure 4: BEECH Predictions for Large Office Buildings in Winter and Summer Based on Commercial End Use Survey Data



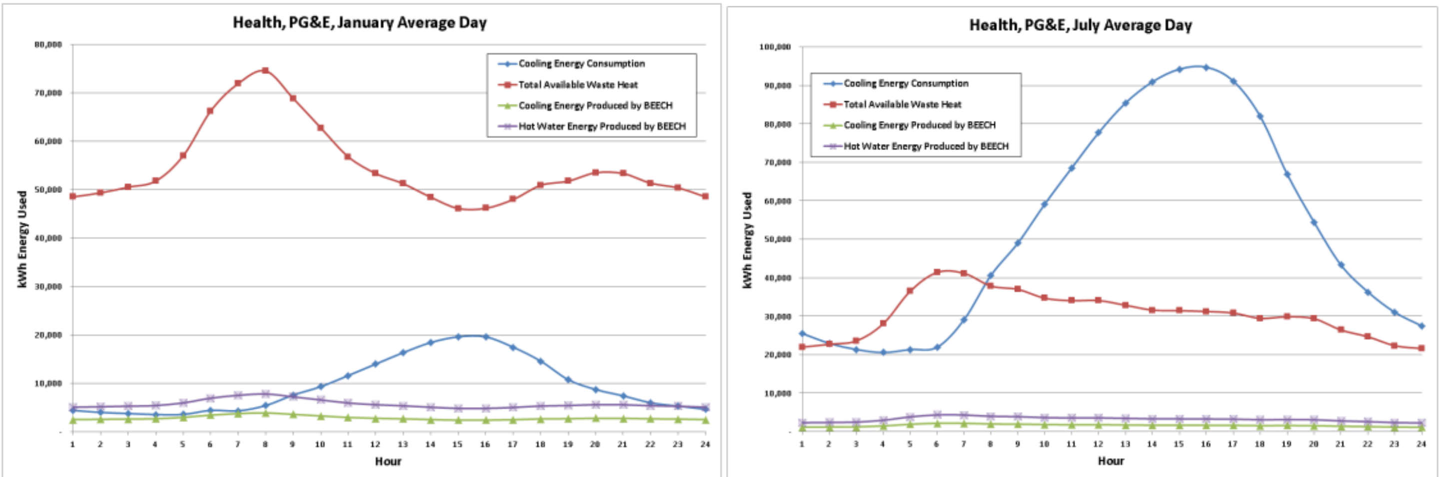
Source: Altex Technologies Corp.

Figure 5: BEECH Predictions for Schools in Winter and Summer Based on Commercial End Use Survey Data



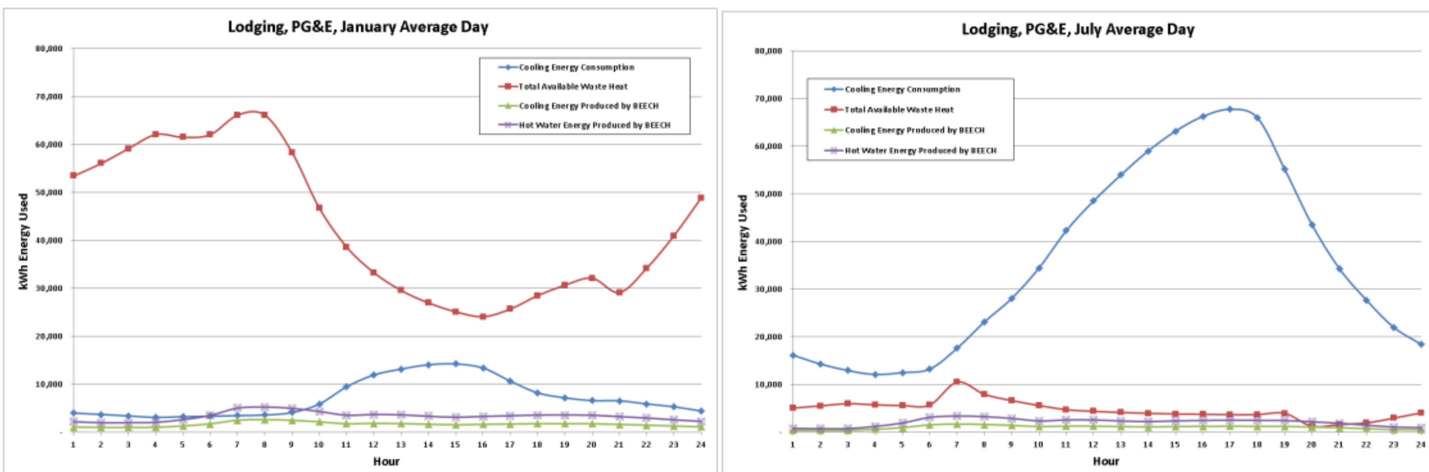
Source: Altex Technologies Corp.

Figure 6: BEECH Predictions for Health Care in Winter and Summer Based on Commercial End Use Survey Data



Source: Altex Technologies Corp.

Figure 7: BEECH Predictions for Lodging in Winter and Summer Based on Commercial End Use Survey Data



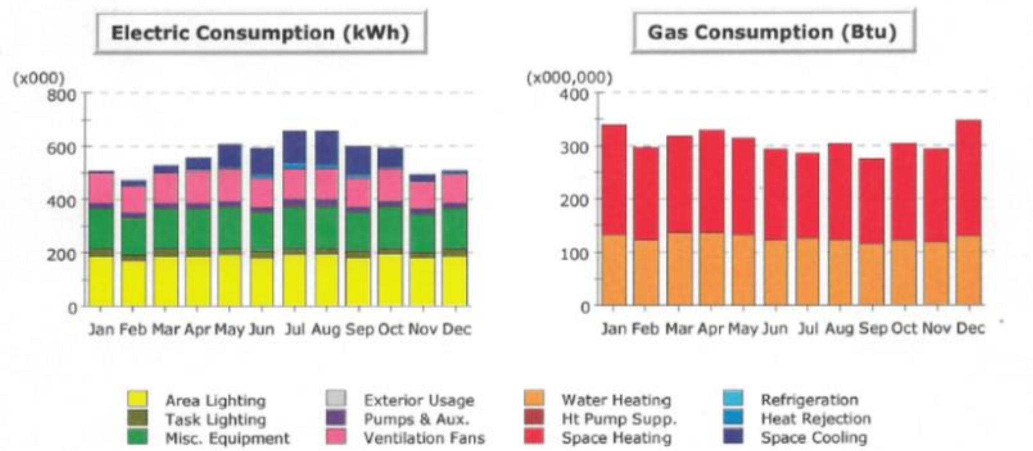
Source: Altex Technologies Corp.

The sectors of large office buildings, schools, lodging, and health care facilities all showed coincident cooling and heating demands, and in proper proportions to allow continuous waste heat BEECH operation. In the cases shown in Figures 4 through 7, predicted BEECH cooling production never exceeds the cooling demand, indicating a high potential for continuous BEECH operation, which would minimize payback time. Other building categories (not shown), such as restaurants, did not have the requisite day-long demands.

Further inferences from these aggregate data sets proved difficult, particularly related to proper system capacity sizing. Many buildings that are too small to be waste heat BEECH candidates (due to low available waste heat) are included in the data. A few large buildings could skew a sector's data set by indicating a day-long heating demand, while in reality a substantial segment of the market has only intermittent demand. A building-by-building approach was needed, to provide more accurate assessments.

The building simulation tool E-Quest was used briefly (Figure 8) to confirm that a single building's usage was consistent with the overall population.

Figure 8: E-Quest Simulation Results: Single Large Office Building Energy Consumption



Electric Consumption (kWh x000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	13.3	22.1	30.5	47.6	82.3	99.8	126.3	124.9	109.0	71.4	27.5	14.1	768.9
Heat Reject.	0.3	1.8	2.6	4.8	9.9	12.3	15.8	15.4	13.1	8.4	1.9	0.5	86.6
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	-	-	-	-	-	-	-	-	-	-	-	-	-
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	-	-	-	-	-	-	-	-	-	-	-	-	-
Vent. Fans	111.7	101.6	112.9	117.7	118.4	107.0	119.8	118.2	107.8	118.6	101.6	110.9	1,346.3
Pumps & Aux.	22.9	20.9	23.1	24.6	25.7	24.2	27.1	27.2	24.7	25.3	21.1	22.9	289.7
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	148.6	134.3	148.6	150.3	152.5	142.5	152.5	152.5	142.5	152.5	138.6	148.6	1,763.9
Task Lights	25.3	22.9	25.3	25.2	25.7	24.3	25.7	25.7	24.3	25.7	23.9	25.3	299.5
Area Lights	188.2	170.1	188.2	188.1	191.8	181.0	191.8	191.8	181.0	191.8	177.5	188.2	2,229.8
Total	510.3	473.7	531.2	558.4	606.2	591.1	658.9	655.7	602.5	593.7	492.2	510.6	6,784.6

Gas Consumption (Btu x000,000)

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space Cool	-	-	-	-	-	-	-	-	-	-	-	-	-
Heat Reject.	-	-	-	-	-	-	-	-	-	-	-	-	-
Refrigeration	-	-	-	-	-	-	-	-	-	-	-	-	-
Space Heat	206.2	174.7	181.9	195.5	182.6	173.3	160.6	183.9	162.0	181.5	174.0	217.8	2,194.0
HP Supp.	-	-	-	-	-	-	-	-	-	-	-	-	-
Hot Water	133.8	123.1	136.4	134.4	132.5	121.1	123.7	121.2	114.6	123.1	118.6	129.8	1,512.1
Vent. Fans	-	-	-	-	-	-	-	-	-	-	-	-	-
Pumps & Aux.	-	-	-	-	-	-	-	-	-	-	-	-	-
Ext. Usage	-	-	-	-	-	-	-	-	-	-	-	-	-
Misc. Equip.	-	-	-	-	-	-	-	-	-	-	-	-	-
Task Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Area Lights	-	-	-	-	-	-	-	-	-	-	-	-	-
Total	340.0	297.8	318.3	329.9	315.1	294.5	284.2	305.1	276.6	304.5	292.6	347.5	3,706.1

Primary Assumptions: Large office, modern construction, 120,000 square feet , distributed on 4 floors; MVC-type chiller with chilled water air handlers, central hot water heating

Source: eQuest; www.doe2.com/equest

These results from a single simulated large office building were still aggregated, though by month. A year-round hot water demand is seen. This includes the heating of city/ground water up to the temperature at which it is consumed, as well as the energy required to maintain a storage tank at a set temperature. Surprisingly, the space heating thermal input only varied +/- 10 percent during the year. This is at odds with the CEUS data, referring specifically to the Large Office category. Both were drawn from the Sempra Energy IOU region, but the variety of potential options in E-Quest makes it difficult to create a truly "typical" building without substantially more data about the "typical" installed equipment and average floor space of the buildings in each segment. One useful conclusion can be drawn from this simulation: the simulation predicts a year-round demand for cooling and hot water, which is a key to BEECH implementation, in either the solar or waste heat incarnation.

At this juncture in the project, the team chose a different approach to matching BEECH output to facility demand. CEUS and E-Quest data both indicated that cooling demand and waste heat would be available year round (and for the waste heat case, throughout the day), and so were unlikely to be limiting factors in the process design. Attention then turned to the other useful output: hot water. If BEECH can provide a substantial offset to a facility's hot water demand by using city water as the cooling medium for one or more heat exchangers, then either the solar thermal or waste heat energy can be recovered by the system at the highest possible efficiency.

Across the full variety of buildings, some potential installations could have insufficient hot water demand to support a cooling output of useful magnitude. This does not automatically disqualify them as potential BEECH installation sites. For these facilities, the process components (for example, the types of condensers) could be specified differently, to better balance the heat recovery, the cooling and the hot water production, but at some overall reduction in efficiency from the ideal case. The better situation would be to install BEECH in buildings of a certain minimum hot water demand. The determination of that demand is described next.

Hot Water Demand Prediction

Required Temperature

The required temperatures for water heating vary depending on application. Figure 9 summarizes this range, which extends to 194˚F (90˚C) for dish rinsing applications. Many different schemes are used to achieve the various temperatures, including multiple boilers operating at different temperatures. It is more common to have multiple steam/water heat exchangers for various points of use in the facility (allowing the kitchen to receive hotter water, and bathrooms and guest rooms to receive lower). An office building without food service would likely have a single boiler and one temperature facility-wide. Since the bacteria that causes Legionnaire's Disease can multiply in stagnant water less than 115 ˚F (46 ˚C), the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) recommends a minimum hot water storage temperature of 140˚F (60˚C). For purposes of BEECH, either the 140˚F or 194˚F limits are appropriate targets for the produced water temperature. The average (167˚F, or 75˚C) will be used in this analysis. This can be achieved using either solar or waste heat BEECH. The energy then required to heat one gallon of water from groundwater temperature, about 65˚F, is 850.7 Btu. This value is used in all calculations.

Figure 9: Representative Hot Water Temperatures

Use	Temperature, °C
Lavatory	
Hand washing	40
Shaving	45
Showers and tubs	43
Therapeutic baths	35
Commercial or institutional laundry, based on fabric	up to 82
Residential dish washing and laundry	60
Surgical scrubbing	43
Commercial spray-type dish washing[a]	
Single- or multiple-tank hood or rack type	
Wash	65 minimum
Final rinse	82 to 90
Single-tank conveyor type	
Wash	71 minimum
Final rinse	82 to 90
Single-tank rack or door type	
Single-temperature wash and rinse	74 minimum
Chemical sanitizing types[b]	60
Multiple-tank conveyor type	
Wash	65 minimum
Pumped rinse	71 minimum
Final rinse	82 to 90
Chemical sanitizing glass washer	
Wash	60
Rinse	24 minimum

[a]As required by NSF. [b]See manufacturer for actual temperature required.

Source: ASHRAE 2003 Handbook

Water Usage – Daily Patterns

Although simultaneous demand for cooling and heating is clear from the large data sets discussed in previously, BEECH will work most efficiently when there is a consistent hot water demand. A continuous, consistent demand is ideal, but installation of a storage system would allow some intermittent operation, if space is available in the facility. ASHRAE standards recommend various storage capacities for various structures (so most facilities will already have some storage capacity). Per ASHRAE, seven liters/occupant is the most common guideline for buildings with bathing facilities. This would likely need to be increased for BEECH to be installed in small facilities with highly variable demand.

Ideally, an hour-by-hour demand curve would have been used as the data source for the hot water consumption (Figure 10).

Figure 10: Residential Average Hourly Hot Water Usage

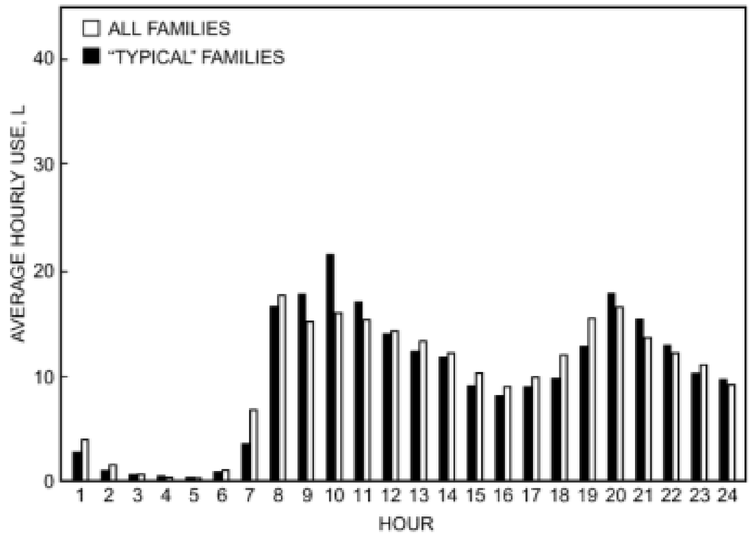


Fig. 10 Residential Average Hourly Hot-Water Use

Source: ASHRAE 2003 Handbook

The E-Quest software was exercised to attempt to derive this data, but it provided only month-by-month data, and the hot water consumption (actual gallons consumed, not just the energy required to heat and/or maintain temperature) could not be accurately derived from the total gas consumption. Instead, the industry-accepted water system sizing guidelines, as provided in the 2003 ASHRAE Handbook, were used. As a starting point, the handbook contains hourly data for some building types and illustrates typical demand for residential applications. As expected, there is minimal overnight demand for hot water. To retrofit the central heating plant in a large apartment complex with BEECH, properly-sized storages could accumulate hot water at night, using the waste heat from the boiler or furnace, and then supply the hot water during the quick rise in demand in the morning. For the solar case, the heat supply and water demand is better matched, and the storage system could supply water for the ongoing demand between sundown and midnight.

Figure 11 provides an alternative data set of hot water demand, for another specific building classification—a full service hotel. Demand is clearly more irregular than the residential case, with major peaks coinciding with morning laundry and guest showers; lunch time; and dinner/check-in. In the overnight and afternoon periods, demand is 20 percent of average.[1] This variable demand is usually met by a hot water or steam boiler with thermal modulation capacity (as well as on/off thermal input at times of very low demand). Overall boiler duty, and therefore, waste heat production, is more uniform, due mostly to the more-uniform demand for

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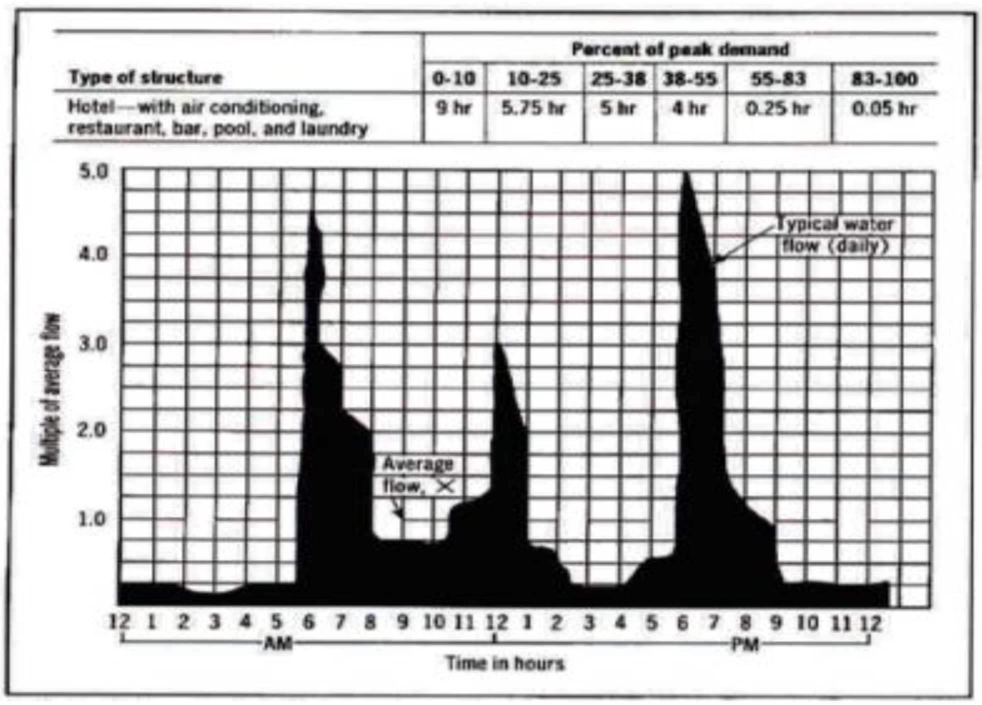
[1] Though this 1989 data might at first seem outdated in light of the water conservation measures implemented in the last 25 years, particularly in California, the curve is actually normalized to average demand. So, if conservation measures have been applied relatively across-the-board with regard to various devices (shower heads, dishwashers etc.), the overall trend is likely still accurate.

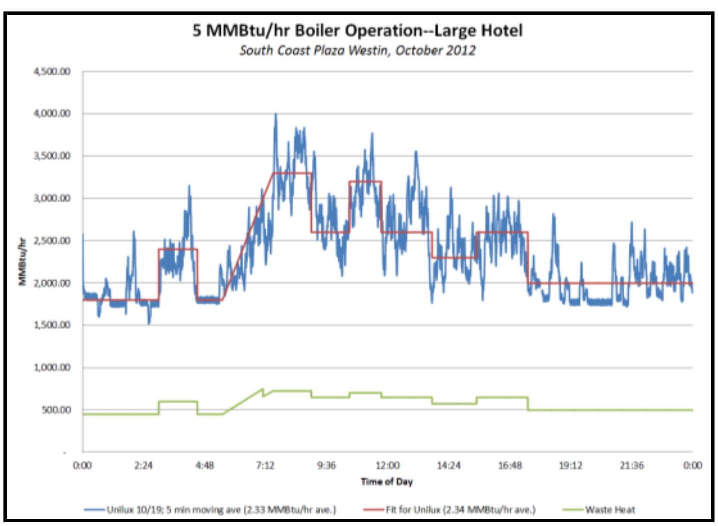
space heat. The modulation ability of the boiler burner and the system storage capacity also contribute. The result is illustrated in Figure 12, where actual measured boiler thermal input from a full-service premium hotel is shown. The early morning and mid-day spikes can be seen, but overall thermal demand at this site was more uniform than in the previous figure.

Figure 11: Expected Hotel Hot Water Demand



Source: Heating/Piping/Air Conditioning October 1989

Figure 12: Hotel Total Thermal Demand



Source: Altex Technologies Corp

Clearly, BEECH must be sized such that the hot water produced is substantially less than the maximum facility demand. The *ideal* BEECH production rate would be the average demand rate, accompanied by *perfectly* sized storage tanks. This is not possible, since demand will vary with season and day. For example, the overall facility thermal demand at the hotel depicted is 10-30 percent lower on Sunday nights than on mid-week nights, due to occupancy patterns. Therefore, oversized storage tanks would be required, and additional heat input would be required to maintain their temperature. As a compromise, if BEECH hot water production is equal to half of the facility average demand, BEECH would satisfy the majority of the demand at all but peak times, without requiring excessively large storage tanks.

Sector-by-Sector Hot Water Consumption

Since global data sets were problematic, specific California facilities with publically-available data were chosen for analysis. The team screened the boiler inventory of the South Coast Air Quality Management District (SCAQMD) for representative facilities in multiple building sectors. The Permits to Operate for the major thermal equipment, as covered by SCAQMD Rule 1146, were obtained online, to verify the maximum potential thermal equipment. Minor sources permitted under Rule 222 were not included. The building "capacities" (usually related to occupancy) were sourced from the corporate websites or online real estate statistics. The selected facilities are shown in Table 1. The team then used the ASHRAE-predicted hot water consumption (Figure 13) to estimate the demand at each building.

Table 1: Sample Facilities for BEECH Installation

Facility	Facility Type	Size	Major Thermal Equipment	Total Max Firing Rate (MMBtu/hr)	Equipment Excepted--Closed Loop
La Serna High School	Education	2850 students	Ajax boiler	2.5	Pool boiler 1.75MMBtu/hr input
Crown Plaza Redondo Beach	Hotel	342 rooms	Rite boiler	3	
Valley Presbyterian Hospital	Large Hospital	250 beds	1 Kewanee steam boiler	10.5	3 Hydronic Boilers 2.5MMB/hr ea
Terranea Resort	Hotel, Resort, Spa	360 hotel rooms	3 C-B Boilers	16.5	
Hollywood Entertainment Plaza	Office Building	165,200 sq ft	ThermoPak Watertube	2.34	

Determined from SCAQMD Boiler Inventory and Public Records

Source: Altex Technologies Corp

Guidelines for hospitals and hotels are not included since their consumption is slightly more complicated to estimate, since the number of hot-water consuming fixtures will vary by the breadth of service offered by the facility. ASHRAE recommends the Hunter Fixture Units Method, but extensive facility information is required to perform the Hunter calculation, including a fixture-by-fixture account of all hot water points of use. For example, a hospital with a large number of therapeutic baths will consume more hot water than one without.

Figure 13: Hot-Water Demands and Use for Various Types of Buildings

Type of Building	Maximum Hourly	Maximum Daily	Average Daily
Men's dormitories	14.4 L/student	83.3 L/student	49.7 L/student
Women's dormitories	19 L/student	100 L/student	46.6 L/student
Motels: Number of units[a]			
20 or less	23 L/unit	132.6 L/unit	75.8 L/unit
60	20 L/unit	94.8 L/unit	53.1 L/unit
100 or more	15 L/unit	56.8 L/unit	37.9 L/unit
Nursing homes	17 L/bed	114 L/bed	69.7 L/bed
Office buildings	1.5 L/person	7.6 L/person	3.8 L/person
Food service establishments:			
Type A—full meal restaurants and cafeterias	5.7 L/max meals/h	41.7 L/max meals/day	9.1 L/average meals/day[b]
Type B—drive-ins, grilles, luncheonettes, sandwich and snack shops	2.6 L/max meals/h	22.7 L/max meals/day	2.6 L/average meals/day[b]
Apartment houses: Number of apartments			
20 or less	45.5 L/apartment	303.2 L/apartment	159.2 L/apartment
50	37.9 L/apartment	276.7 L/apartment	151.6 L/apartment
75	32.2 L/apartment	250 L/apartment	144 L/apartment
100	26.5 L/apartment	227.4 L/apartment	140.2 L/apartment
200 or more	19 L/apartment	195 L/apartment	132.7 L/apartment
Elementary schools	2.3 L/student	5.7 L/student	2.3 L/student[b]
Junior and senior high schools	3.8 L/student	13.6 L/student	6.8 L/student[b]

[a]Interpolate for intermediate values. [b]Per day of operation.

Source: ASHRAE 2003 Handbook

However, other sources[2] estimate 35 gallons per day per occupant in hospitals and 20-35 gallons per day per occupant of hotels. Each room in these facilities can be assumed to have 1.2 occupants.[3] So, in lieu of the fixture inventory, these simpler estimates were used. Consumption at the Crown Plaza was then estimated to be 27.5 gallons per occupant per day (average of the 20-35 gpd range), and consumption at the Terranea was assumed to be 35 gallons per occupant per day, since it is a full service resort. For purposes of this study, a 90 percent occupancy rate was assumed.

To evaluate an office building, the number of occupants must be known, and can be derived from the square footage and typical densities for modern office space. Allowing 200 square feet per employee,[4] the Hollywood office building is assumed to support 826 working people, for a daily demand of 3,139 liters, or 826 gallons. A thermal efficiency was also assigned to each facility, based on the type and age of boiler—the older Kewanee and Cleaver-Brooks boilers installed at Valley and Terranea were assigned 78 percent thermal efficiencies, and the smaller or newer boilers were assigned 80 percent efficiencies. These assumptions are fairly conservative, since most boilers achieve their maximum thermal efficiency at higher firing rates, but operate most often at some partial capacity. Table 2 presents the results of the analysis.

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[2] Engineering Toolbox Hot Water Consumption Per Occupant. http://www.engineeringtoolbox.com/hot-water-consumption-person-d_91.html . Accessed February 23, 2014.

[3] Lehr, Valentine. *Hot Water Requirements for Hotels.* Heating/Piping/Air Conditioning, October 1989.

[4] Miller, Norm. *Estimating Office Space per Worker*. Burnham-Moores Center for Real Estate, University of San Diego. May 2012.

With the exception of the Crown Plaza, the hot water thermal demand is approximately 10 percent of the max potential waste heat. This relationship may be somewhat misleading, since the installed thermal capacity does not indicate the true thermal demand. The Westin hotel had, until recently, two 10 MMBtu/hr boilers, but an average thermal demand of fewer than 3 MMBtu/hr or less for nine months of the year.

Clearly, the school and the large office buildings are projected to have very low overall hot water use, either averaged across the entire day, or across the likely times of high usage (school or business hours). These two sites would be poor choices for a BEECH installation, as there is little demand for the hot water. The school, in particular, would use even less of the potential thermal output during summer vacation. However, as noted above, a version of BEECH that used air-cooled condensers could be considered, though the overall system thermal efficiency would be decreased.

Table 2: Hot-Water Demands and Use for Various Types of Buildings

Facility	Assumed Thermal Efficiency	Max Available Waste Heat	Ave. Hot Water Demand	Ave. Hot Water Demand	Alternate Demand Calculation
		Btu/hr	*Gpm*	*Btu/hr*	*Gpm*
La Serna High School	80%	500,000	0.85	42,500	2.0 (during 10 hr school day)
Crown Plaza Redondo Beach	80%	600,000	7.0	357,300	
Valley Presbyterian Hospital	78%	2,310,000	5.4	275,600	
Terranea Resort	78%	3,630,000	9.5	484,900	
Hollywood Entertainment Plaza	80%	468,000	0.57	29,278	1.37 (during 10 hr work day)

Source: Altex Technologies Corp

The other facilities have substantially more demand, and are better fits for BEECH. Usage profiles and storage capacities will vary, but if the production is targeted at 50 percent of average, as discussed above, the system target should be 2.7 to 4.75 gallons per minute (gpm), with 3.7 gpm being the nominal.

One important conclusion of this analysis is that the 360,000 Btu/hr heating goal of BEECH should not be met solely by heating city water to storage temperature. Since the system hot water output needs to be less than average demand, 360,000 Btu/hr would produce 7 gpm of 167˚F (75˚C) water, which is more than could be used by 4 out of 5 sites analyzed. Instead, the lower average water flow rate noted above should be targeted for the typical BEECH system, with the understanding that if a facility with high hot water use wanted to install BEECH, another water-cooled condenser could be used instead of an air cooled condenser, thus producing more hot water and improving overall system thermal efficiency.

The above facility analysis used the known boiler thermal capacity for calculating the available waste heat. A similar analysis can be performed for the solar thermal case, with available installation surface area serving as the "capacity" reference. As described in the following section, a 3.7 gpm hot water/5 tons cooling system requires 250,000 Btu/hr thermal input to generate peak output. Using the Kingspan Thermomax collector as the reference device, each 30-tube collector panel can generate a peak output of 10,000 Btu/hr.

Therefore, at least 25 panels would be required, under optimum conditions. Allowing some margin for production of maximum output at non-peak insolation times, a 30 collector array can be specified using Kingspan's recommended spacing for a 30˚ collector angle on a flat roof or parking structure,[5] which is appropriate for a central California location. Five rows of six panels would occupy 2,730 square feet. This footprint could be accommodated by any of the listed sites, either on the building roof, as shown in Figure 14, or over adjacent parking structures. Façade-mounting options are also available for these panels. In this case, no shading allowance is required, and the 30 panels would occupy only 1,480 square feet, also a reasonable area.

Figure 14: Kingspan Thermomax Collector Installation Example

Source: www.kingpsan.com

Calculating Cooling Output with Hot Water Production

The BEECH system will include two condensers that could use water as the cooling medium: the power cycle condenser and the refrigeration cycle condenser. The system could also include a heat exchanger, which would extract more heat from the oil or glycol, prior to returning it to the heat source. The condensers could also be air cooled, but water is preferable, since it allows lower condensing temperatures during hot ambient air conditions, and will increase system thermal efficiency. Altex engineers modelled the BEECH system with CHEMCAD process modeling software, and it was relatively simple to change HX types and cooling media in this

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[5] *Complete Solar Solutions.* Kingspan Solar. Undated copy, p. 128. Also available at http://www.kingspansolarmanual.com.

model. Various configurations and combinations of the HX were evaluated, with a target of producing 3.7 gpm of 167˚F (75˚C) water.

The operating temperature of a condenser is set by the pressure at which condensation is occurring. In BEECH, the condenser pressures (and therefore, temperatures) are limited by the pressure ratios at which the expander and compressor devices can operate efficiently. Neither condenser will be operated above 167˚F (75˚C), so the oil/water HX (shown as HX-3 in Figure 1) is required for production of 167˚F (75˚C) water. The condensers were evaluated as water or air condensers, both individually and in combination. The refrigeration cycle condenser will operate at a lower temperature/pressure, and therefore is the best candidate for water cooling. Ideally, the water flow path would be sequential, from the AC condenser to the power condenser, but the temperature/pressure conditions required to balance those two cycles always yield an AC condenser water outlet temperature that is too high to be used as the coolant for the power cycle. Instead, the power cycle heat exchanger is air cooled, and the water-to-oil/glycol heat exchanger is included in the design.

Site Specification Summary

Sites for BEECH installation should have the following characteristics:

1. Available heat source, in the form of:
    a. 2800 sq. ft of solar collector installation area, or
    b. Waste heat of >400 ˚F, at a flow rate such that 250,000 Btu/hr can be recovered
2. Cooling demand of at least five tons, with demand coincident to thermal production
3. Hot water demand greater than 3.7 gpm on average
    a. Site should have existing or be able to expand water collection capacity to store water heated at times of low demand—exact capacity will depend on building type and usage pattern
    b. Site will still require a parallel, demand-based hot water heating system to meet variable demand. This is likely to already exist or be planned at any site.

Ideal sites with the following characteristics will allow BEECH to operate at the greatest efficiency, electrical savings, or natural gas use reduction:

1. Minimum hot water demand of 3.7 gpm, preferably at a temperature higher than 167 ˚F
2. If the waste heat source is fired (for example, a boiler), that it is fired on natural gas only (will also improve heat recovery HX durability)
3. The site water storage system can be adapted via—or built with—an internal or external heat exchanger, such that the heated oil can be used to maintain stored water temperature at or above 140 ˚F (replacing any natural gas firing otherwise used for maintaining water temperature).

CHAPTER 3: Expander/Compressor Selection, Design, and Fabrication

Expander/Compressor Technology Selection

The expander and compressor functions are the heart of the BEECH system. For commercial deployment, the expander/compressor unit (or units) will need low initial cost and low maintenance cost. The project proposal used an ejector as an example of an integrated expander/compressor unit that met those criteria. Altex has extensive experience with ejectors, and used steam ejectors in PIR-11-027 to develop a low-cost, steam driven chiller system. However, ejectors sacrifice isentropic efficiency for their simplicity. This might seem to be of less importance in a waste heat or solar application, where the cost of the input energy is very low, but if options with higher efficiency exist, their use will decrease the mass flow of working fluid. This will reduce piping and system size, and also reduce pump and fan operating costs.

Early in the system design and engineering task, Altex staff sought to confirm the best technology for the expander and compressor. They completed a literature survey of various expanders, as well as compressor technologies that could be reversed to become expanders. The results are given in Table 3. Since refrigeration compressors are widely available as low-cost, mass-produced devices, the expander was seen as the more challenging machine. Turbine-type expanders (such as those used in many Organic Rankine power cycles) were rejected early in the process. They can operate at high expansion ratios and are often very compact, but they operate at high rotating speeds and are relatively expensive. Gerotor-style compressors can be reversed to act as expanders, but have low efficiency. Some university research has found >70 percent efficiency using oil-free scroll compressors (such as those used in air compressors) as expanders with refrigerants. However, these units are not lubricated and are designed for 2000-hour lifetimes. This is adequate for the intermittent duty cycles of air compressors, but would not be acceptable for a continuously operating waste heat recovery system.

The most promising, and eventually selected, technology is the refrigerant scroll compressor, a sample of shown in Figure 15. These units are manufactured in large quantities and have a proven track record of reliability in residential, commercial, and industrial environments. They have relatively high efficiencies, and are designed to work with common refrigeration lubricants, avoiding the durability issues of the oil-free compressors. Manufacturer data indicates that their isentropic efficiencies can exceed 70 percent, and various academic researchers have confirmed 68-71 percent efficiency when the compressor's operation is reversed to function as an expander. Finally, the scroll compressor is strong enough to withstand the high expander inlet pressures of the double expansion process. In its designed mode of operation, the same model of scroll compressor may be operated on a variety of refrigerants. Some, such as R-410a, require compressor outlet pressures of >400 pounds per square inch (psia) when operated in hot climates (since the condenser in those locations will have a relatively high pressure and temperature.), which is required for the R-134a/R-1234xx working fluids described in Chapter 4.

Table 3: Expander Efficiency Summary

Source	Refrigerant	Device Type	Data Source	Isentropic Eff.	Notes
Mathais et al 2009	R-123	Gerotor	Experimental	35%	
Single Nozzle Ejector	R-718	Ejector	Manufacturer/ Altex Analysis	44%*	90 psig motive pressure
Multi-Nozzle Ejector	R-718	Ejector	Manufacturer/ Altex Analysis	48%*	90 psig inlet pressure
R.B. Peterson et al. 2008	R-123	Rigid Scroll	Experimental	50%	
Badr et al 1985	R-113	Rotary Vane	Experimental	55%	Cited by Declaye
Zanelli & Favrat (1994)	R-134a	Refrigerant Compressor Scroll	Experimental	63%	Peak efficiency Measured
Saitoh et al. 2007	R-113	Scroll	Experimental	63%	
Ziviani et al (ASME 2012)	R-245fa	Refrigerant Compressor Scroll	Theoretical	68%	Peak efficiency at 5.5 Pr
Declaye et al (Univ. Liege, 2013)	R-123	Air Compressor Scroll	Experimental	68%	Oil free, low durability
Declaye et al (Univ. Liege, 2013)	HFE7000	Air Compressor Scroll	Experimental	68%	Oil free, low durability
Lemort et al (Univ. Liege 2011)	R-245fa	Refrigerant Compressor Scroll	Experimental	71%	Peak efficiency @ 3.75 Pr
Declaye et al (Univ. Liege, 2013)	R-245fa	Air Compressor Scroll	Experimental	71%	Oil free, low durability
Declaye et al (Univ. Liege, 2013)	R-245fa	Air Compressor Scroll	Experimental	76%	Mag coupling, 12bar inlet pressure
Air Squared	R-134a/R-245fa	Custom Scroll	Manufacturer Claim	70-80%	Manufacturer Rating, Prototype

***Ejector efficiencies are expressed as the effective isentropic efficiencies of the expansion or compression actions, for correct comparison to other equipment types which perform those functions in separate devices. These efficiencies were used in CHEMCAD analyses and verified against manufacturer claims.**

From a cost standpoint, a five cooling tons scroll compressor has a retail price (as a service part) of approximately $1,800, including the electric motor, hermetically sealed vessel, and all required mounts and fittings. By removing the retail mark-up and the cost of the electric motor, the scroll components are estimated to cost less than $1,000 to manufacture. The total cost of an expander/compressor unit is therefore expected to cost <$3,000 to manufacture. The two scroll pairs will be coupled via a driveshaft, and will be housed in a hermetically sealed vessel capable of appropriate strength and safety factor for the pressures and temperatures calculated in the CHEMCAD process design.

Figure 15: Sample Scroll Compressor Components

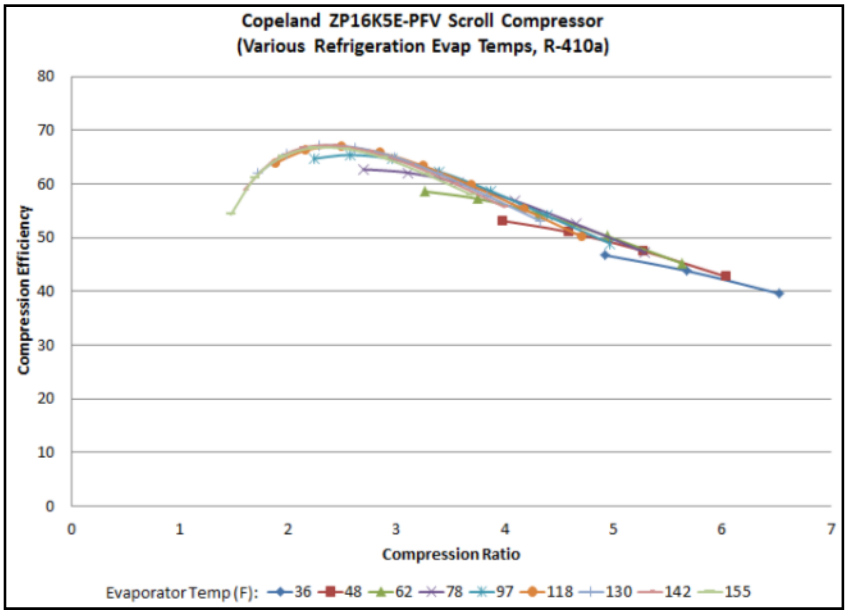
(l-r rotating scroll, fixed scroll)

Source: Altex Technologies Corp.

Expander/Compressor Performance Prediction

Manufacturers of compressors publish efficiency ratings for refrigeration system designers, and are usually expressed in terms of condenser and evaporator temperatures, for a given refrigerant. These reference points do not usually extend to the high temperatures (and therefore, high pressures in the expander) required for waste heat recovery. However, if the data for a different refrigerant, one that has much higher saturation pressures, is analyzed, the expected behavior of the BEECH expander can be determined. Altex engineers analyzed the performance of many scroll compressor models by back-calculating operating pressures and pressure ratios from published data, to understand the operating characteristics. A sample data set, derived from Emerson-Copeland data for a small one-ton compressor (Figure 16). The data shows a strong correlation between a compression ratio and efficiency, regardless of refrigerant temperature. After the CHEMCAD model was created, the Excel-based analyses were not needed, as the expander/compressor curves could be directly entered into CHEMCAD. The scroll units eventually chosen for BEECH have peak efficiency near 3:1 compression (or expansion) ratio, consistent with the Lemort study referenced in Table 3.

Figure 16: Sample Scroll Compressor Performance



Compression Ratio as Indicator of Isentropic Efficiency

Source: Altex Technologies Corp.

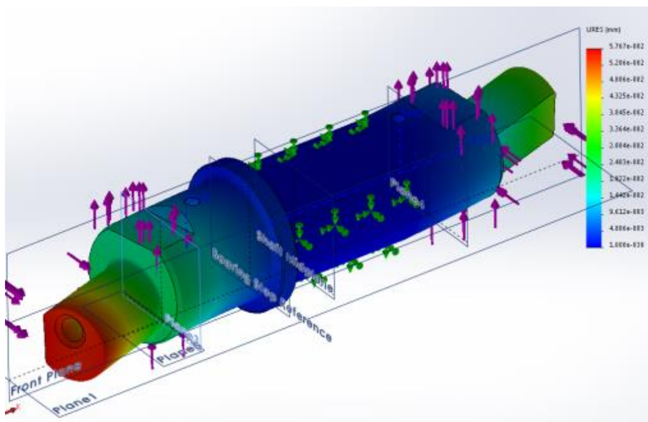
Many modern mechanical vapor compression (MVC) refrigeration systems operating between 9,000 Btu/hr (0.75 refrigeration tons) and 600,000 Btu/hr (50 refrigeration tons) use scroll compressors, and low cost manufacturing techniques for these devices are well-developed. A scroll compressor uses a pair of scrolls to progressively compress a vapor. One scroll is fixed, and the other orbits in an eccentric path, moving the mass of vapor to the smallest volume and highest pressure, which is achieved at the center of the scroll pair.

In a conventional MVC system, the scroll compressor is driven by an electric motor. In BEECH, the compressor is driven by another scroll pair, operating as an expander. The orbiting scroll of the second pair drives a common shaft supported by a center bearing. This important simplification eliminates the cost of the electric generator and motor, as well as their inefficiencies in converting electrical energy to mechanical work.

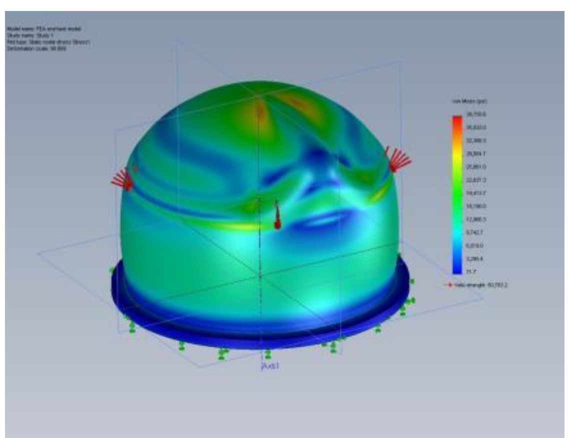
The CHEMCAD study of pressure ratios, mass flows, and system efficiencies yielded a process design with a bypass flow that equalizes the volumetric flow rates to the expander outlet and compressor inlet. This allowed the same model of scroll to be used for both sides of the unit. The team procured two Emerson-Copeland ZB58KCE-TFD units, removed the key scroll components, and recycled the electric motors. A coordinate measuring machine was used to fully dimension one set of scrolls and related hardware, and these dimensions were used to create an integrated expander/compressor model in Solidworks.

Altex engineers designed a custom steel shaft to join the two scroll assemblies, and verified its strength using Solidworks Finite Element Analysis (Figure 17). They also specified a roller bearing pair to support the assembly. Key characteristics of the shaft, including hardness and surface finish, were based on the engineering recommendations of the shaft seal and bearing suppliers. The shaft design also incorporates oiling features similar to that of the refrigeration compressor, but adapted to be used with an external oil pump. In a production version of BEECH, the oil pump would be integrated into the mechanical assembly, but this additional engineering exercise was outside the scope of the current project. The finished shaft, assembled to the bearing pair, is shown in Figure 18.

Figure 17: Sample Finite Element Analysis Results



Expander/Compressor Shaft Deflection Analysis



Tank Head Stress Analysis

Source: Altex Technologies Corp

Figure 18: Machined Shaft and Bearing Assembly

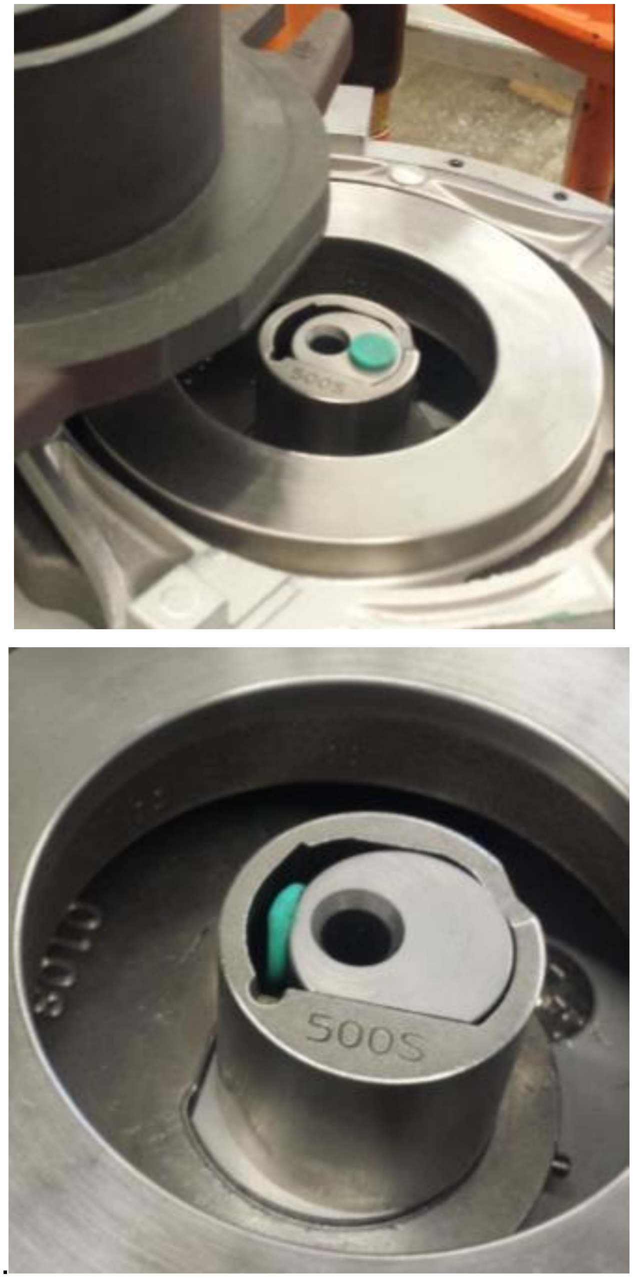
Source: Altex Technologies Corp.

The shaft and bearings are supported by a machined center section. Altex engineers incorporated many of the dimensional features from the scroll compressor, to leverage the proven design of the commercially available unit. Modifications were made to accommodate the external oiling system, as well as inclusion of removable vessels ends. The vessel ends were re-used from the purchased compressors, but modified to permit repeated removal of the refrigerant connection, as well as the heads themselves. Solidworks FEA confirmed the strength of the modified vessel. Double ferrule compression fittings were used for the refrigerant lines (as contrasted to the brazed connections used on the stock compressor), and O-ring-sealed flanges were added to the vessel ends to ensure tight sealing, but allow easy access to the expander/compressor internals.

To verify the mechanical design, an aluminum center section was first machined, and assembled to the scrolls, shafts, bearings, and vessel heads. As-assembled dimensions were checked using calipers, micrometers, and in some cases, clay (Figure 19). Leak tightness was first tested with dry, pressurized nitrogen, and after resolving minor issues, was tested again with a vacuum pump. The unit achieved a vacuum of fewer than 500 microns, which is acceptable by refrigeration industry standards.

Altex engineers then assembled the complete expander/compressor subsystem, including the external oil system and control valves. Leak testing of this assembly quickly identified an issue with the oil pump, which was not designed for the inlet pressures that will be experienced in the BEECH system. This issue was eventually resolved by designing a high-pressure-capable shaft seal, and retrofitting the pump with this seal.

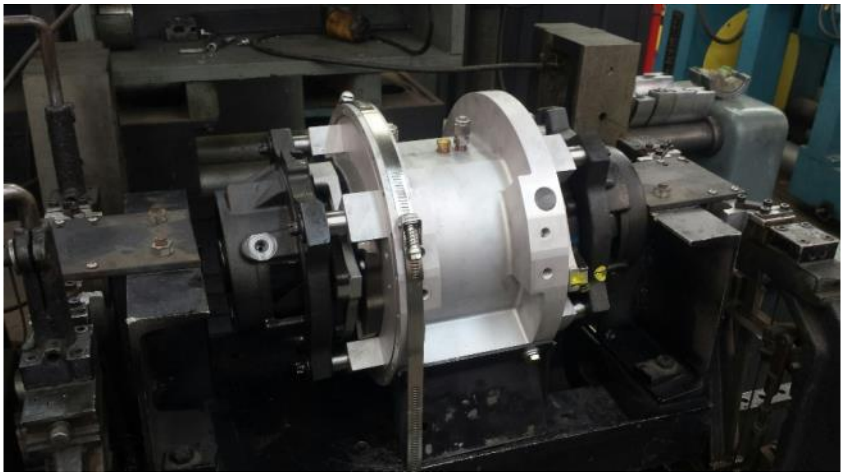
Figure 19: Measuring Expander/Compressor Internal Clearances with Clay



Source: Altex Technologies Corp

After the necessary checks and measurements were completed on the assembly, the center section was then re-machined to create a balancing fixture. The as-manufactured refrigeration compressors include a counterweight to properly balance these systems, but they were no longer properly balanced after the various modifications. A belt-drive pulley was also machined, to interface with a spin-balancing machine, and the expander/compressor was re-assembled, using the modified center housing and the pulley. The complete assembly was installed in the balancing machine (Figure 20), and spun to quantify the imbalance of the assembly. The counterweights were modified until balance was acceptable.

Figure 20: Expander/Compressor During Balancing



Source: Altex Technologies Corp.

Once checks were completed with the aluminum center section, a new steel center section, incorporating the improvements identified with the aluminum unit, was machined, and the oil reservoirs and connections were welded in place (Figure 21).

Figure 21: Steel Center Section, Machined and Welded

Source: Altex Technologies Corp.

After this welding, several secondary machining operations were performed to ready the unit for final assembly. The final assembly process took longer than expected, since the bearings had to be shimmed and modified to create the proper preload (to minimize shaft displacement, yet not cause excess frictional losses.). Once this was resolved, the final assembly was completed, and additional lubrication tests were performed to verify function of the new, high-pressure-capable seals in the oil pump (Figure 22).

Figure 22: Final Expander/Compressor Assembly During Oil System Testing

Source: Altex Technologies Corp.

CHAPTER 4: Working Fluid Selection

Initial Calculations and Screening Criteria

The Altex team pursued working fluid selection activities in parallel with the system process design. For continuity, some of the process design is described in this chapter, and refers to the final process design as previously shown.

As the team evaluated various tradeoffs between expander/compressor efficiency, condenser types, and site hot water demands, it became clear that condenser operating temperatures would govern the overall thermodynamic processes, and the chosen working fluid's properties would have to be favorable at those conditions. The temperature of the working fluid in the refrigerant condenser was set at ~75˚F (23.9 ˚C), based on California groundwater temperatures, and a reasonable heat exchanger approach temperature. The temperature then sets the condenser pressure, based on the saturation properties of the chosen fluid.

Altex modelled several system arrangements that incorporated a water cooled condenser in the power cycle, with water flowing in series or in parallel to the refrigeration cycle condenser, but these either increased water flow above facility needs, or did not facilitate the ideal 3:1 expander pressure ratio. As a result, the team eventually specified an air-cooled condenser for the power cycle, and air temperature set the design temperatures and pressures of the power cycle.

Given the above guidelines for pressures, the fluid screening process proceeded in parallel with the process design, and various working fluids were used in the CHEMCAD analyses. The team reviewed more than thirty available refrigerants, using the following primary criteria:

- Low Global Warming Potential (GWP) and Ozone Depletion Potential (ODP)
- No planned phase-out
- Ability to operate sub-critically at the projected expander inlet temperatures/pressures (which would vary fluid-to-fluid). While super-critical operation is not intrinsically problematic for some fluids, very little supercritical material property data has been published for the fluids whose pressure/temperature characteristics were compatible with available scroll compressors. This would have made accurate modeling very difficult.
- Preferably non- explosive and non-toxic for use in an experimental system

Many available refrigerants were rejected due to high GWP or ODP. Refrigerants formulated for low temperature refrigeration applications often failed the sub-critical criteria. Explosive (for example, isopropane) and potentially toxic (for example, ammonia) refrigerants were also considered, since they are gaining market share in Europe, and some equipment is available in the United States, but CHEMCAD analyses showed that their operating properties at high temperatures did not create an efficient power cycle, and so were not deemed to be worth the risk.

The most common working fluid for Organic Rankine power cycles is R-245fa, and much of the preliminary CHEMCAD analysis was performed with R-245fa, since it passed all the screening criteria above, although the GWP is 950. Its ability to operate at reasonable pressures (<500 psig) at the temperatures required for heat recovery or solar thermal makes it an obvious choice for these applications. However, for a single working fluid system, the chosen fluid must also meet system requirements in the refrigeration cycle. Unfortunately, the density of R-245fa is very low at typical refrigeration evaporator temperatures. Table 4 illustrates this, and compares R-245fa to R-134a, which is commonly used in automotive and some chiller applications.

Table 4: Working Fluid Comparison, Sample Conditions

Property*	R-245fa	R-134a
Refrigeration Cycle Evaporator Temperature	50 °F	50 °F
Refrigeration Cycle Evaporator Outlet/ Compressor Inlet Pressure	9.5 psia	50 psia
Fluid density	0.239 lbm/ft^3	1.027 lbm/ft^3
Power Cycle Expander Inlet Pressure (9x Compressor Inlet Pressure)	85.5 psia	450 psia
Power Cycle Expander Inlet Temperature	205.5 °F	240.1 °F
Fluid Specific Volume	1.789 ft^3/lbm	8.563 ft^3/lbm

***Simplified analysis, does not include piping/minor component pressure drops; compressor inlet temperature assumes 10°F superheat; expander inlet temperature assumes 50 °F superheat.**

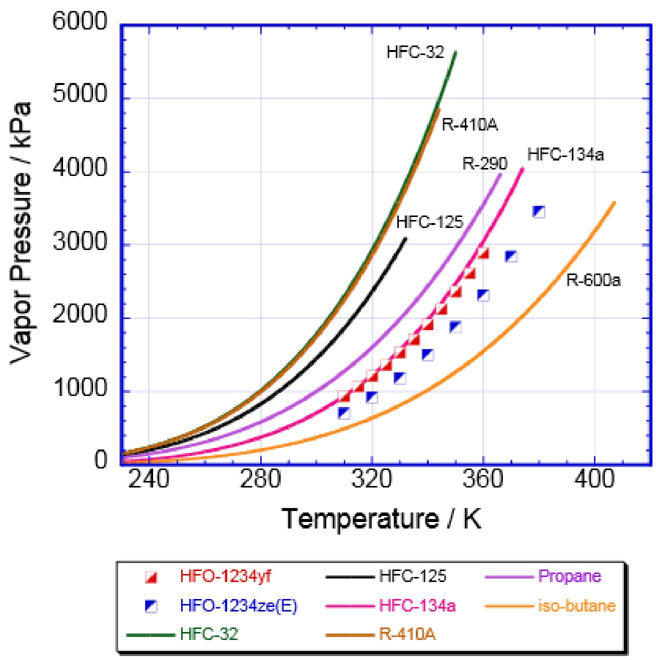
Source: Altex Technologies Corp.

The density of the R-245fa, particularly at the low temperature, low pressure condition of the refrigeration cycle implies that a large volumetric flow will be required. The 7.5x change in density, from the expander inlet to the compressor inlet, implies that piping in the R-245fa refrigeration cycle would need to be much larger than in its power cycle. More importantly, the compressor volumetric capacity would have to be very large. The preferred design direction for the process and for the expander/compressor was to use a single scroll pair of each, joined by a common shaft. The two units would then rotate at the same speed; for a given scroll and shaft speed, the units would operate with constant (but not necessarily identical) volumetric displacements. The great difference in volumetric flow would have made the compressor scroll relatively very large. So large, in fact, that no commercially-available scroll compressors could have worked for a 60,000 Btu/hr cooling output. A complicated gearbox or a planetary, magnetic coupling could have joined multiple compressors to one expander power shaft, but this would defeat the goal of a low-cost expander/compressor. Another alternative considered was to select multiple oversized expanders and operate them inefficiently. Even for the waste heat case, this was shown to be undesirable in CHEMCAD analyses, since the increase in pumping costs outweighed the value of the cooling. It would also be possible to draw excess power from the expander shafts to drive electric generators (creating a combined heating, cooling, and power system), but this alternative would add substantial cost to the system.

Selection of R-1234xx, as Approximated by R-134a

As previously shown, R-134a is more than four times as dense as R-245fa at the temperatures of interest in the BEECH system, implying much more reasonable volumetric flow rates and smaller component sizes. R-134a was initially screened out of the BEECH selection process since it has a GWP of 1300 and as such did not meet the screening criteria for low GWP and phase out. However, Honeywell and other manufacturers are developing low GWP/ODP replacements for R-134a. Honeywell's R-1234 family of refrigerants will have a GWP of 4-6, which is non-zero, but still much better than R-134a's 1300. Variants of R-1234 are already in production for other applications, and their use is expanding into refrigeration. Complete material properties are not available for these fluids, but initial studies (Figure 23) show they will be very similar to R-134a, particularly the R-1234yf variant. After the team became aware of these developing refrigerants, they used R-134a in CHEMCAD analyses. Since R-134a is commercially-available and has many components and oils designed for compatible use, it decreased risk in the build of the BEECH system under this project. The long-term goal for the BEECH technology is to use these advanced refrigerants with low GWP/ODP; the changes required to convert will be minimal, since R-1234 is intended to be a direct replacement for R-134a, with little or no efficiency decrease.

Figure 23: Fluid Comparison: R-1234xx and Currently Available Refrigerants



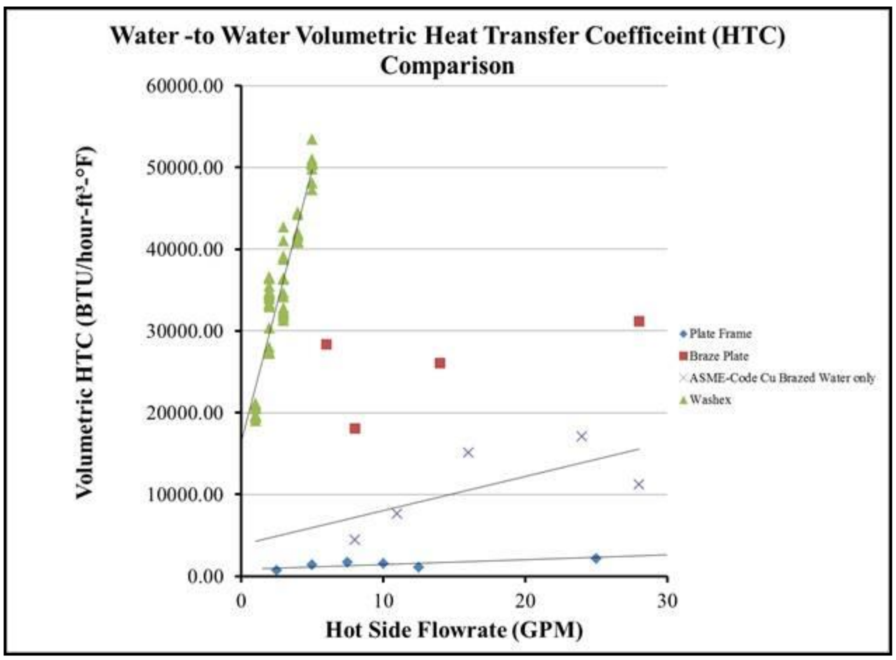
Source: Higashi, Yukihiro. Thermophysical Properties of HFO-1234yf and HFO-1234ze.

CHAPTER 5: Generator Heat Exchanger

Generator Design

The generator—which would be similar in function and performance specifications for the solar and waste heat cases—was based on Altex's novel minichannel heat exchanger technology. Altex has previously demonstrated high volumetric heat transfer coefficients from these units (green data points in Figure 24), and their fabrication cost is estimated to be 30-50 percent cheaper than chemically-etched and diffusion-bonded minichannel units presently available.

Figure 24: Previous Altex Mini-channel Heat Exchanger Performance Compared to Existing Technologies

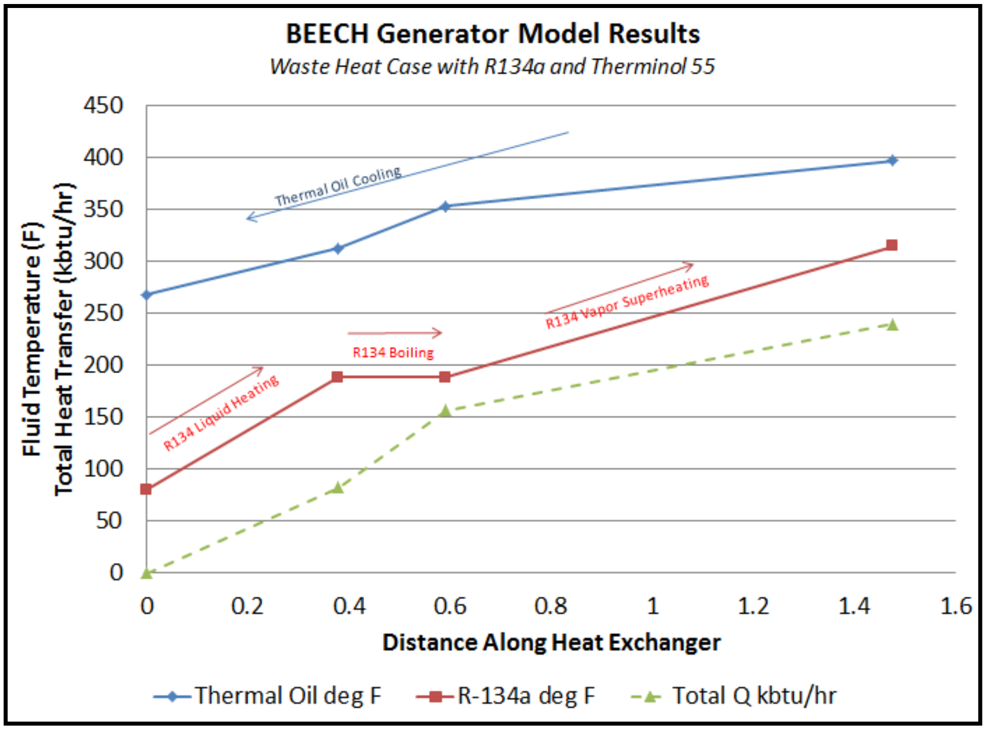


Source: Altex Technologies Corp.

For the BEECH project, Altex chose its High Efficiency, Low Cost (HELC) minichannel technology, which has a high volumetric heat transfer coefficient and high pressure capability. Altex engineers updated the existing minichannel heat exchanger model to use R-134a, and either 50/50 ethylene glycol/water mix (for solar thermal) or Therminol 55 (for waste heat). The previous heat exchanger model was designed for single phase flow, and had to be updated not only for the evaporation behavior of R-134a, but also its variable specific heat.

Figure 25 shows the predicted generator performance, and clearly shows the liquid heating, vaporization, and superheating phases of the process occurring inside the generator.

Figure 25: Generator Heat Transfer and Temperature Change

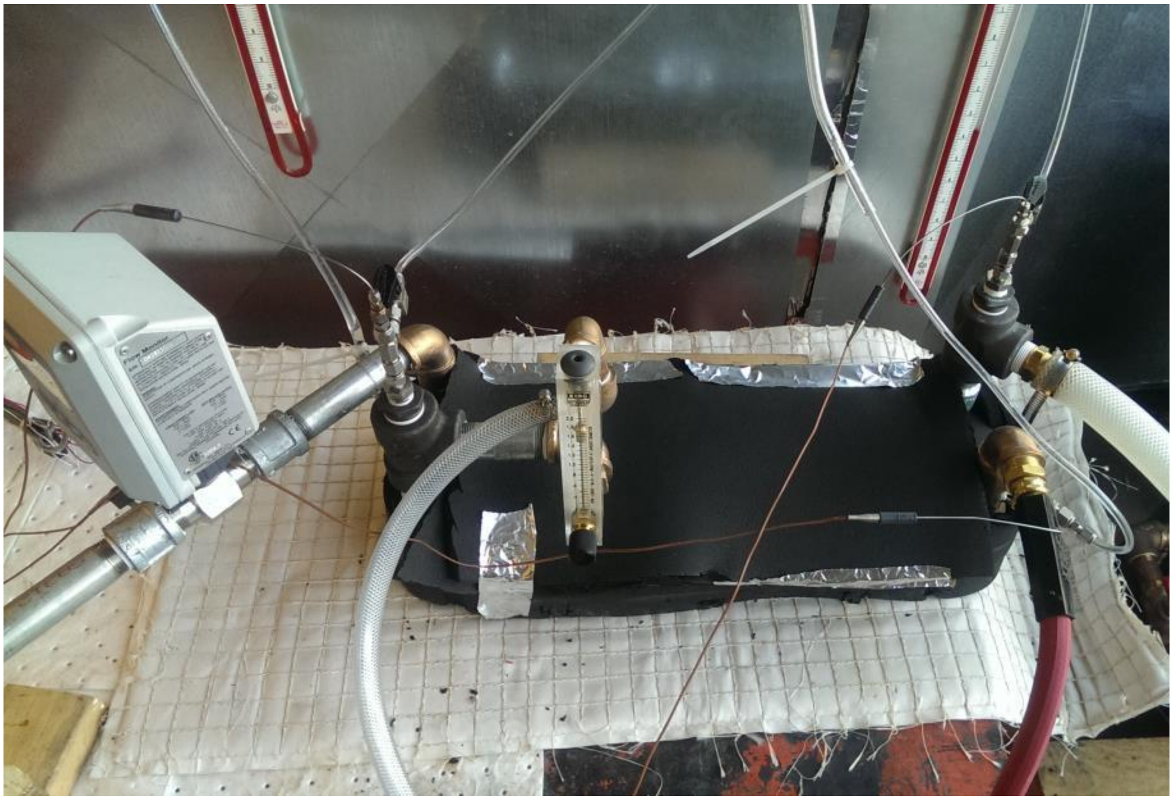


Source: Altex Technologies Corp.

Working under match funds provided by the Department of Energy, Altex and its fabrication partners designed, machined, and brazed a high-pressure capable, minichannel heat exchanger with brazed-on end plates. The geometry and construction details are not identical to the BEECH HELC design, but the brazing and assembly procedures, as well as subsequent testing, provided valuable inputs to the BEECH plans and reduced risk and cost for the BEECH unit. The test article was successfully brazed and passed helium leak check at the supplier facility (Figure 26). At Altex, it underwent heat transfer and pressure drop testing, and then successfully passed pressure testing up to 3000 psi, which indicates a factor of safety of more than 6.0 for the BEECH system requirements.

Based on the model and laboratory test results, Altex engineers created a mechanical design of the HELC generator. The unit is sized specifically for the BEECH system, and has endplates optimized for strength and weight, based on the maximum design pressure of waste-heat-driven BEECH. Altex engineers created mechanical drawings for all subcomponents, and supervised the fabrication and quality check activities. Vacuum Process Engineering (VPE) of Sacramento, CA, was a minor subcontractor to this activity, and performed additional quality check services, as well as industry-leading brazing processes for the generator. VPE is also partially supporting the BEECH effort on a match–funds basis.

Figure 26: High Pressure Heat Exchanger, Insulated During Heat Transfer Testing

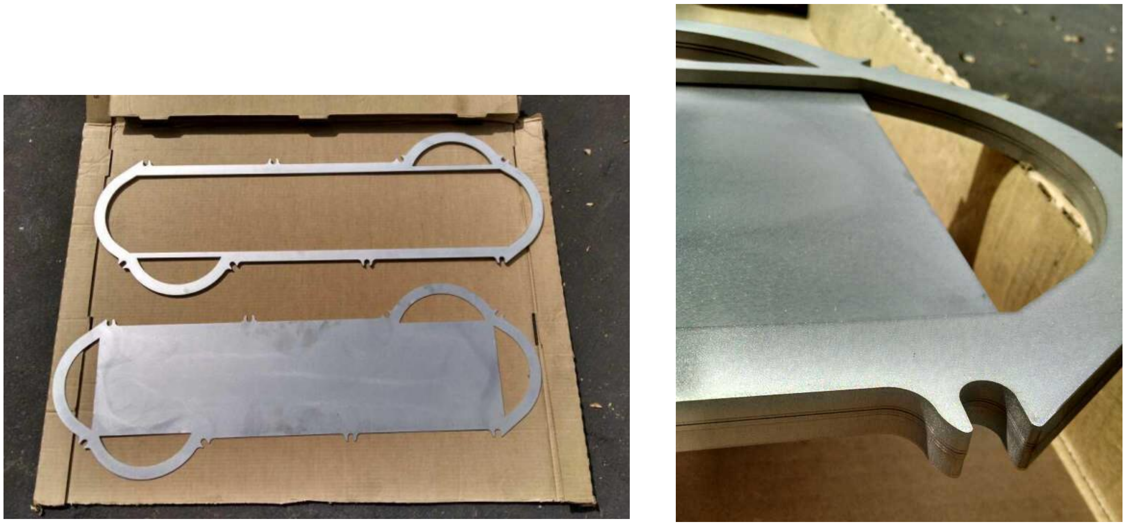


Source: Altex Technologies Corp.

As shown in Figure 27 and Figure 28, Altex engineers then designed and oversaw the fabrication of brazing trial components, and VPE led manufacturing process developments to create several coupons and subscale test articles that refined the brazing process parameters required to create the novel heat exchangers.

While the brazing development was ongoing, Altex cooperated with minor subcontractor Legacy Chiller Systems to procure a conventional brazed-plate type heat exchanger, manufactured by Alfa Laval. Shakedown and performance testing was performed using this generator. A full scale HELC was produced near the end of the project, but too late to be tested in the waste heat BEECH system. Ongoing tests of this unit are instead being performed on a match funds basis, under support from the United States Department of Energy.

Figure 27: Generator Brazing Trial Components



a) Plates and frames

Source: Altex Technologies Corp

Figure 28: Generator Brazing Trial Assembly

Source: Altex Technologies Corp.

CHAPTER 6:
Heat Input: Heat Recovery Heat Exchanger or Solar Thermal Collectors

Heat Recovery Heat Exchanger

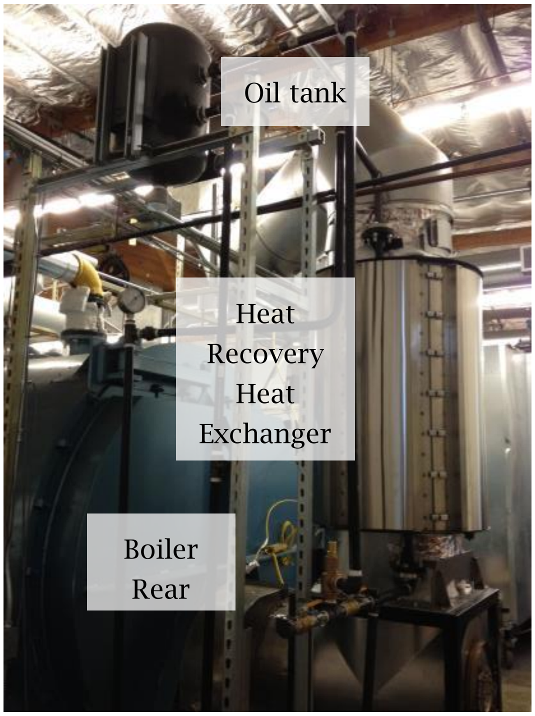
The heat recovery heat exchanger (HRHX), also known as an “economizer” in the boiler industry, transfers heat from the hot waste stream to a working fluid, which is then pumped to the generator. It is theoretically possible to directly vaporize the refrigerant in the HRHX, but the high pressure refrigerant would require a much more robust and expensive heat exchanger, able to withstand the high temperatures of the waste heat, as well as the high pressures of the refrigerant. Economizers are well-known to the boiler industry and are already designed and rated for boiler service. They are commonly used to pre-heat boiler water or returned condensate from the steam system, thus improving boiler system thermal efficiency.

Altex engineers developed a specification for the HRHX and contacted several manufacturers for quotation. After receiving bids and technical information, Cain Industries was chosen as the supplier. Cain offered two different economizer designs, and the cylindrical version was chosen. The heat transfer performance at three potential operating temperatures was calculated by Cain, and Altex engineers updated the CHEMCAD model to account for these parameters.

The HRHX is essentially an off-the-shelf design from Cain, though the inlet/outlet connections were specified as welded, flanged connections (instead of their usual pipe-thread connections), to provide improved sealing when operated with thermal oil. The resulting unit has a maximum operating temperature rating of 750 ˚F (399 ˚C).

Altex engineers designed a support structure to adapt the HRHX to the existing boiler in the test facility, and also specified an oil pump that met the pressure and flow requirement as calculated in the updated CHEMCAD model. As shown in Figure 29 and Figure 30, the economizer was installed in the Altex facility, and appropriately sized piping was installed per the P&ID, to connect the thermal oil to the BEECH system. Figure 31 shows the oil pump installed in the facility, and piped to the economizer.

Figure 29: Heat Recovery Heat Exchanger Received at Altex

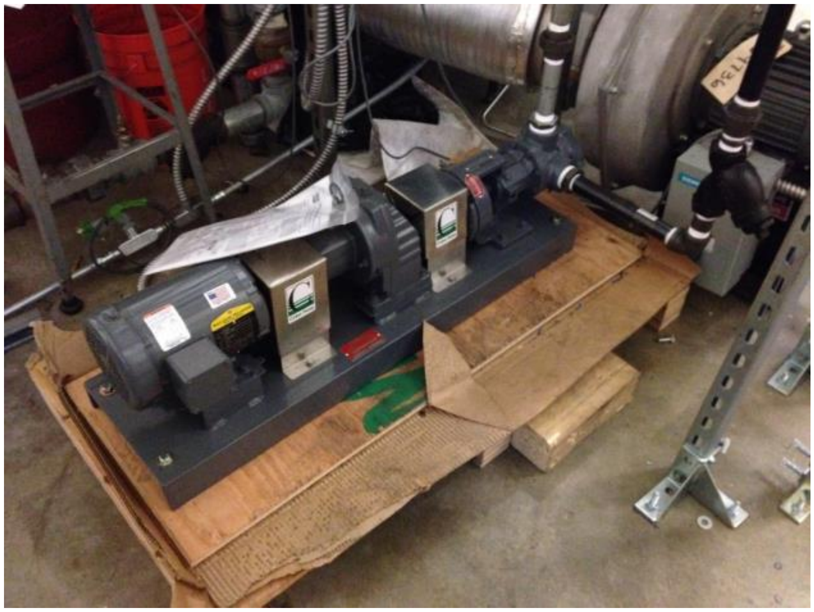


Source: Altex Technologies Corp.

Figure 30: Heat Recovery Heat Exchanger and Oil Tank Installed

Source: Altex Technologies Corp.

Figure 31: Oil Pump In Place and Piped



Source: Altex Technologies Corp.

Solar Collectors

Rooftop solar collectors transfer energy from solar insolation to a working fluid. The simplest, and cheapest, collectors are used for pool heating and are of plastic construction. As temperature and pressure capability increase, so does cost. Since higher expander inlet temperatures in BEECH will lead to higher system efficiencies, high-temperature-capable collectors were the obvious targets for testing. Ideally, solar and waste heat BEECH would use the same working fluid to transport heat from the solar collector to the generator. Due to the >500˚F (260˚C) potential of waste heat, a synthetic oil will be used for that application. Oil could also be used for the solar case, but Altex was open to other working fluids, such as glycol, particularly if supporting equipment was commercially available and proven.

Altex engineers also specified and sought quotations for a solar thermal collector from three manufacturers: ergSol, Chromasun, and Kingspan. For various reasons, as described in the Task 3 report, Chromasun and ergSol were not good fits. Kingspan Solar offers several type of solar collectors, including an evacuated-tube version that is capable of the pressures and temperatures required for BEECH, when operating with pressurized glycol. Kingspan also distributes pump stations, expansion tanks, and glycol blends specially formulated for solar thermal use. Their San Francisco-based staff were able to provide details on installation for the unique BEECH application, and could supply efficiency data beyond standard published data, to cover the extended temperature range of BEECH. Altex purchased their largest single-panel unit for use in the Task 3 Solar and Subcomponent testing. The assembly has 30 evacuated tubes and a maximum thermal output of 10,000 Btu/hr, under standard Solar Rating and Certification Corporation (SRCC) conditions. A larger, multi-collector array was considered, but was too large for temporary installation at the Altex facility. At the time of the decision, the expander/compressor technology and necessary thermal or refrigerant capacity had not been determined, and so size matching of collector and expander/compressor was not attempted.

Since the subcomponent test apparatus would be a temporary installation, the standard Kingspan roof or façade mount options were not useful. Altex engineers designed an adjustable aluminum frame that included ballast trays for cement blocks. Altex engineers analyzed the frame for strength and stability using Finite Element Analysis, and then oversaw its fabrication by Nunez Precision Welding in Milpitas, California. The assembled collector and frame is shown in Figure 32.

Figure 32: Solar Thermal Panel Installed at Altex

Source: Altex Technologies Corp.

CHAPTER 7:
Refrigerant Pump Selection

The refrigerant pump[6] increases the pressure of the liquid refrigerant prior to boiling and superheat in the generator. Altex engineers researched pumps that were capable of meeting the specifications for the solar and waste heat versions of BEECH. This meant that the pump would need to produce the 450 psig outlet pressure and tolerate 300°F fluid temperature required for the waste heat application. The flow rates of the subcomponent test and the full waste heat system are substantially different, but ideally, the same manufacturer and product line would offer pumps capable of both flow rates. The results of the preliminary sourcing activities are summarized in Table 5.

Table 5: Pump Selection Matrix

		Pressure Ability	Flow Rate Ability		
Brand	Pump Type	450 psi capable	Solar (Subcomponent)	Waste Heat (Full System)	Comments
Hermetic-pumpen	Gear	No	No	No	Minimal R134a pumps, low flow, max 142 psi
Cornell Pump	Various	Yes	No	No	Excess flow rate, low boost pressure
Weir (Wemco/Roto-Jet)	Various	Yes	No	No	Not compatible with refrigerant
MTH Pump	Gear	No	No	No	Max Rated Differential pressure is 125psi
Taco	Various	No	No	No	No refrigerants
NR Products	Gear	Yes	No	No	Low pressure only, not rated for continuous duty
Hy-save	Centrifugal/mag	No	Yes	Yes	Max Temp - 161F, Low boost pressure
Micropump	Turbine	Yes	No	No	Max Rated Differential pressure is 125psi
Uraca	Piston	No	No	No	Very high flow; very high pressure only
Fluid O Tech	Gear	No	No	No	low pressure only
Zenith/Colfax Pump	Gear	Yes	No	No	low viscocity
Ceme/Ulka	Solenoid	Unknown	Unknown	Unknown	RFI sent, no response
Hydracell	Diaphragm	Yes	Yes	Yes	Leading candidate, selected as source for Task 3

Source: Altex Technologies Corp.

The Hydracell pump was the only unit capable of meeting the high pressure and low flow rates that matched the single solar panel's thermal output (Figure 33). The Hydracell unit is a diaphragm-type metering pump capable of pumping the very-low viscosity refrigerant.

The range of Hydracell models cover the flow needs of both subcomponent and waste heat test systems. The pump, as initially mounted in the test setup, is shown in Figure 34.

As described in Chapter 8, the diaphragm pump did not work well in solar testing, and so Altex engineers sent twelve additional RFQ's to manufacturers and distributors for a different pump to be used in the waste heat system. A pump from Speck was selected. Unlike the diaphragm pump, the Speck unit is a multi-stage, side channel pump and is specifically rated for refrigeration service.

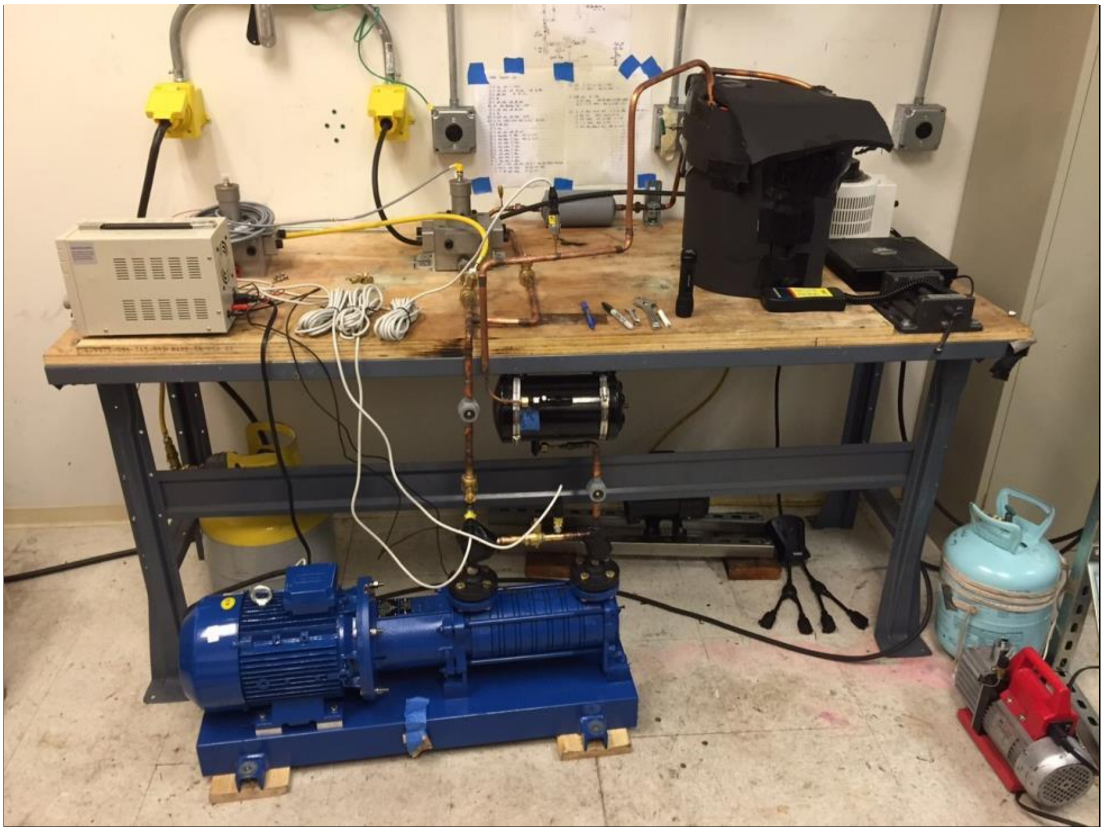
[6] Labelled in Figure 1 as PM-2.

Figure 33: Hydracell Refrigerant Metering Pump

Source: Altex Technologies Corp

Altex engineers, with the assistance of minor subcontractor Oxford Engineering, rebuilt the subcomponent test apparatus to accommodate the larger flow of the full-capacity Speck pump. The team tested the pump at a variety of outlet pressures and rotational speeds. Testing was completed in August 2015, and the pump demonstrated much better performance than the diaphragm pump.

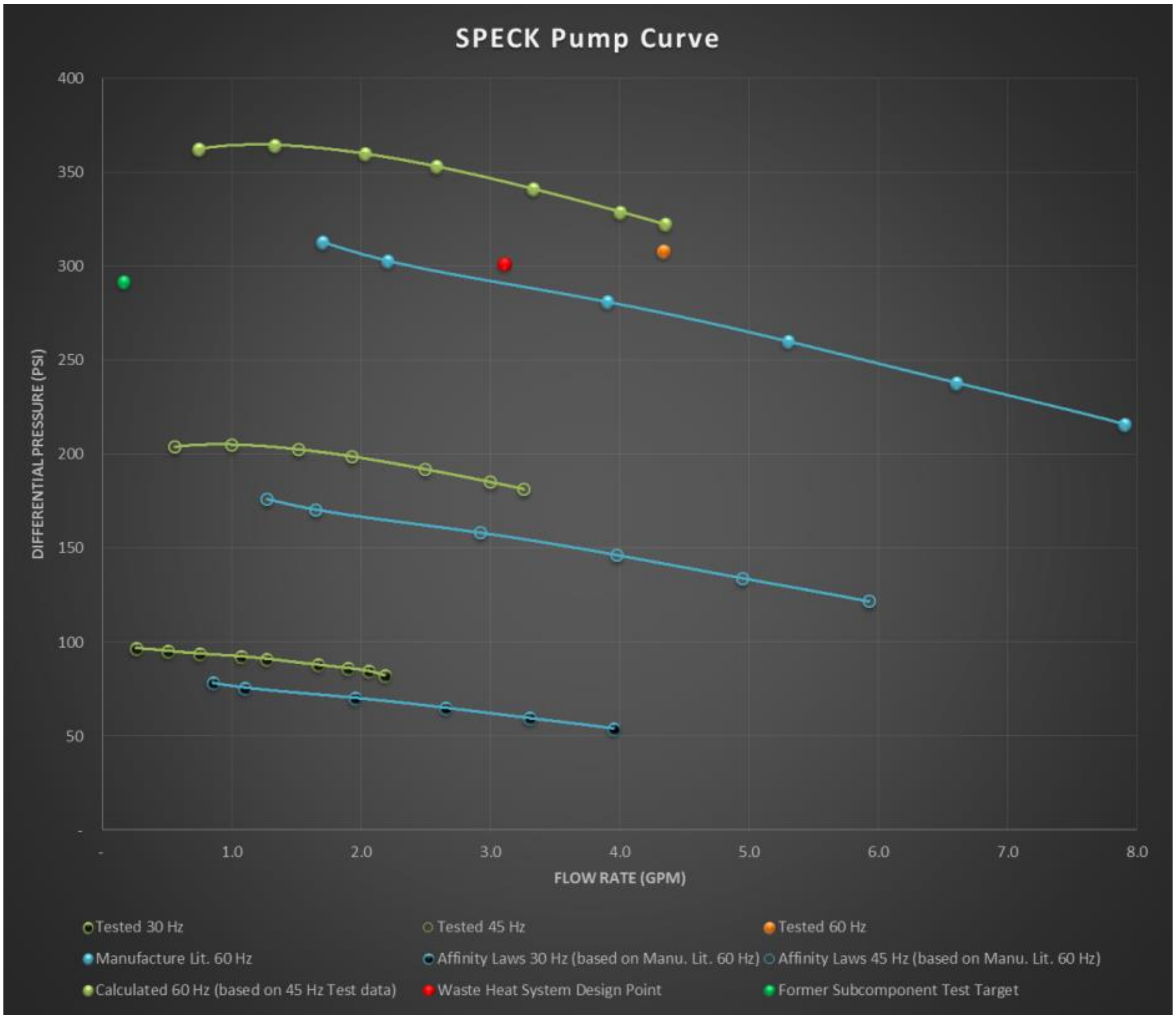
Figure 34: Subcomponent Test Set-up, Rebuilt with Speck Refrigerant Pump and Water Bath for Refrigerant Cooling



Source: Altex Technologies Corp.

The pump achieved the required flow at the required rate and pressure, and the rate could be varied using a variable frequency drive, which is consistent with the planned operation of both the waste heat and solar BEECH systems. The chosen model of pump was sized to support the process conditions of waste heat BEECH, which has a designed operating flow of 3.1 gpm at 301 psi differential pressure. As shown in Figure 35, the pump achieved 4.34 gpm at 308 psi differential pressure, when operated at 60 Hz speed. Overall, performance was slightly better than predicted by the manufacturer's literature, perhaps due to different environmental conditions, or differences in the material properties (for example, viscosity) of R-134a, as compared to the fluid used by the manufacturer in their rating procedure. Further testing also demonstrated that the refrigerant flow rate could also be decreased by lowering pump speed, as shown by the 30 and 45 Hz curves shown in Figure 35. After the successful test, the pump was assembled into the BEECH system, using brackets designed to align the pump with the existing BEECH plumbing.

Figure 35: Refrigerant Pump Test Results



Source: Altex Technologies Corp.

CHAPTER 8: Solar/Subcomponent Testing

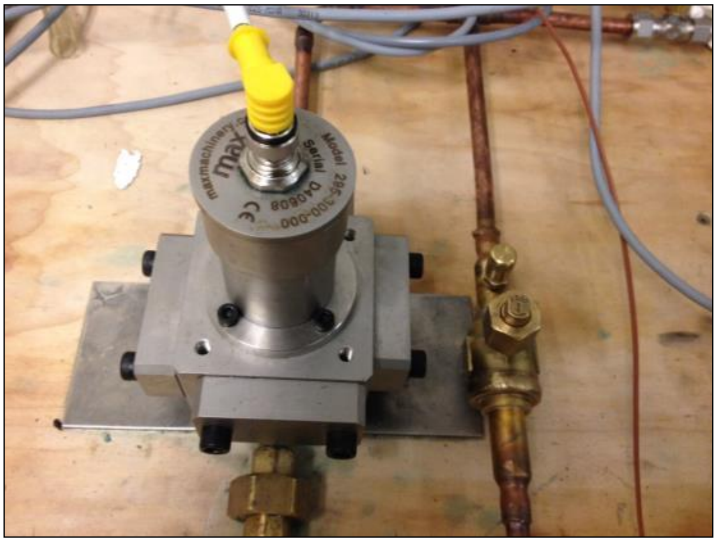
The key components of solar BEECH are the solar collector, the novel expander/compressor unit, the generator, and the refrigerant pump. The other heat exchangers, pumps, and valves are expected to be commercial, off-the-shelf parts. A goal of this testing was to gain experience with these standard components in preparation for the full scale system buildup and operation. For example, the refrigerant and water flowmeters that will be used in the full scale system were purchased in advance and used in the solar testing, to verify their accuracy and repeatability. Many other parts procured for the solar subcomponent tests (such as sight glasses, tanks, and valves) were similar in type and design to those that were used in the larger waste heat system.

Ideally, all four of the key components would have been tested during the subcomponent tests. However, the expander/compressor could not be sized to be consistent with the available and installable solar collectors, and so a manual expansion valve was used instead, to provide the maximum amount of control over the expansion ratio. Delays in executing one of the minor subcontracts delayed fabrication of the custom generator until after subcomponent testing was complete. However, a heat exchanger with similar channel sizes and materials of construction (brazed stainless steel) was sourced, and provided adequate performance in this testing.

System Assembly

To evaluate the thermal performance and efficiency of the BEECH system, the flow rates of the various fluids must be measured. At elevated temperatures, both glycol and R-134a have low viscosities, which are not compatible with some flowmeters (for example, paddlewheel type). For example, R-134a, measured at the bypass line, has a dynamic viscosity of 0.171 Centipoise (cP); water, at standard temperature and pressure, has a dynamic viscosity of 0.899 cP. Fortunately, piston-type flowmeters are minimally affected by viscosity, as long as the pistons can seal in the presence of the working fluid. Altex purchased Max Machinery's 213 and 214 series flow meters based on their ability to measure low flow while maintaining high accuracy and high resolution throughout the testing ranges. The units are also rated up to 437˚F (225˚C) and are available with SAE O-ring fittings, which are compatible with refrigeration systems. Both flowmeters were purchased from Max (located in Healdsburg, California). For subcomponent testing, the 213 measured refrigerant flow and the 214 measured glycol flow. In the full-sized system, the 214 will measure total refrigerant flow, and the 213 will be used to measure either bypass or refrigeration cycle refrigerant flow. The 213 meter is shown in Figure 36, as installed in the solar test apparatus.

Figure 36: Max Machinery Refrigerant Flowmeter



Source: Altex Technologies Corp

To measure water flow, Altex selected Proteus-brand water flow meters (Figure 37). Altex has used Proteus meters on previous projects with good success. Two meters were purchased directly from Proteus in Mountain View, California. For subcomponent testing, one was used to measure cooling water supply to the condenser.

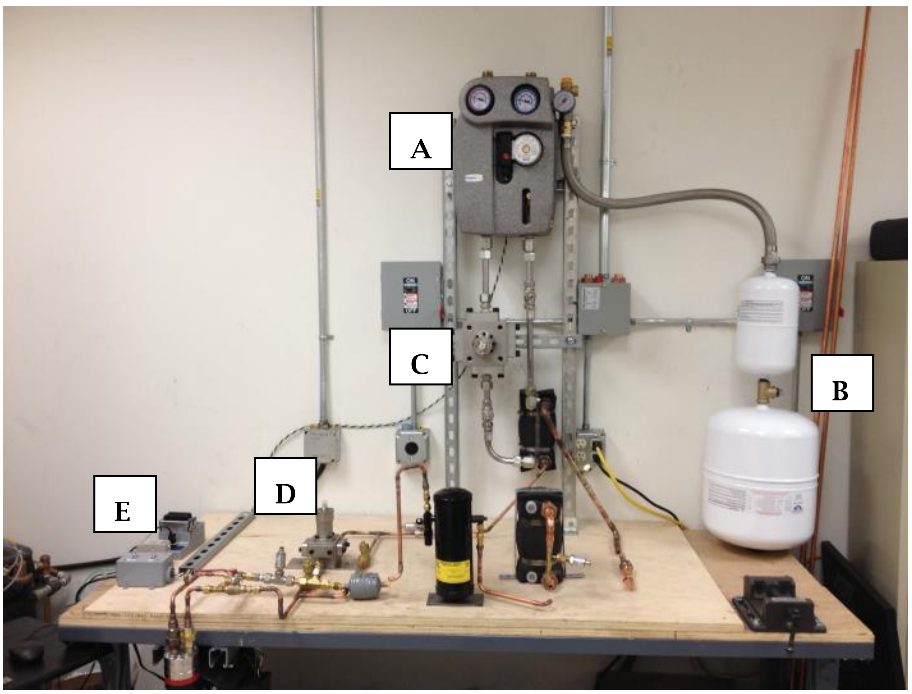
Figure 37: Proteus Water Flowmeters

Source: Altex Technologies Corp.

After the test set-up was built (Figure 38), the Altex test engineer leak tested the system. The glycol/solar circuit was tested with compressed air, and then charged with glycol. Kingspan

Solar staff made an on-site visit to Altex to assist with this process. The glycol side of the system remained charged and leak-free throughout testing. The test engineer then pressure tested the refrigerant circuit with compressed nitrogen (at 500 psig). After eliminating all leaks, the engineer evacuated the system with a vacuum pump, per standard HVAC practice. After satisfactory vacuum was achieved—indicating no leaks and that all water had been evaporated and discharged from the system—the engineer charged the system with R-134a refrigerant.

Figure 38: Solar/Subcomponent Test System Lab Set-up



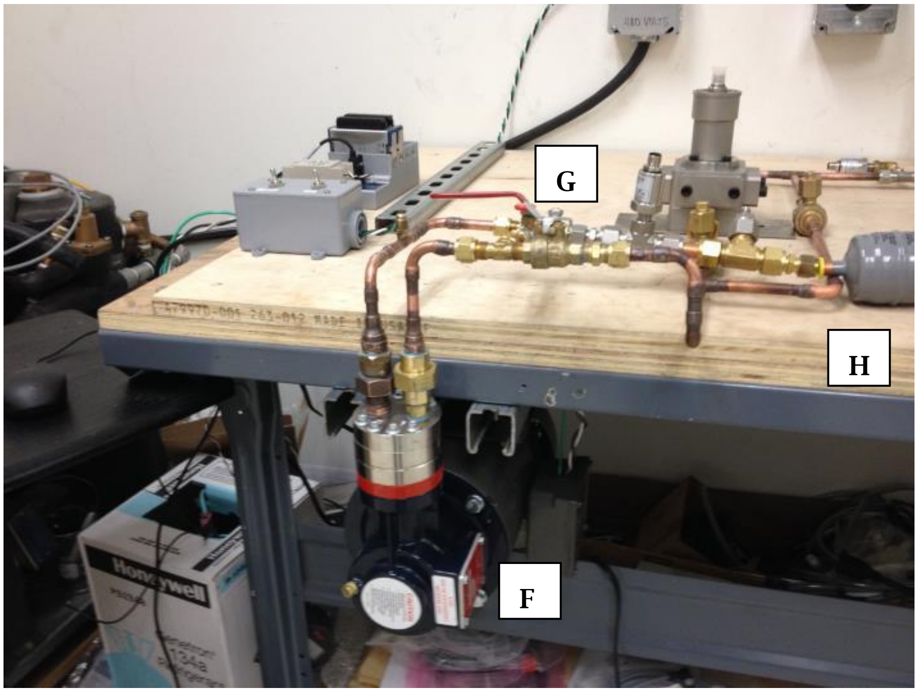
A: Pump Station; B: Expansion Tanks; C: Glycol flowmeter; D: Refrigerant flowmeter; E: Connection to Data Acquisition System.

Source: Altex Technologies Corp.

A process of troubleshooting and component upgrades then began (Figure 39). The high pressure regulator that was chosen as an expander replacement was found to have been configured incorrectly at the factory, and had external vent and check valve functions that caused refrigerant leaks and prevented refrigerant flow at high pressures, respectively. The valve was first rebuilt using the correct parts (provided at no cost by the manufacturer due to the factory error) for vent-free operation, and then was replaced with an adjustable needle valve at the end of testing, to allow more precise control during start-up. The needle valve required adjustment during testing, but was straightforward in operation.

Refrigerant pump priming became the major testing roadblock after engineers achieved a leak-tight system. Upon start of the diaphragm pump, no flow was measured by the refrigerant flow meter. The pump manufacturer had warned that the small-displacement pump was prone to vapor lock, and might have to be operated for some period of time at start-up to clear any evaporated refrigerant. The test engineer tried this, but the pump heated up, thus increasing

Figure 39: Solar System Lab Set-up, Pump Detail



F: Refrigerant Pump; G: Start-up Bypass Plumbing; H: Filter/Dryer

Source: Altex Technologies Corp.

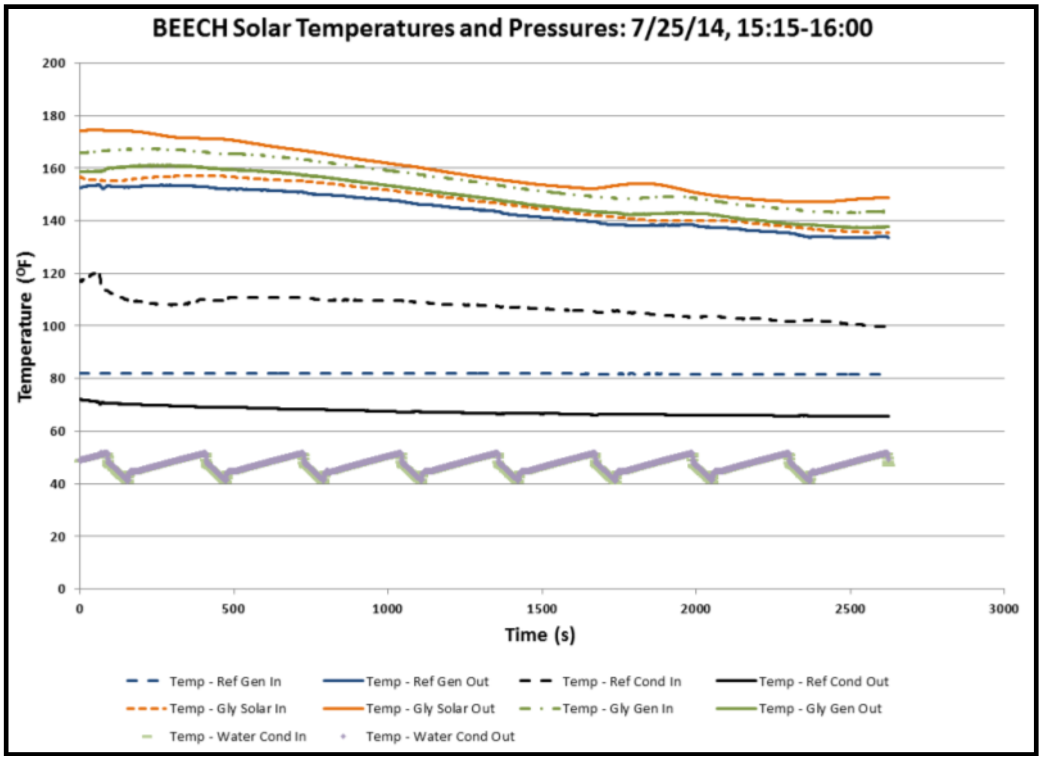
the likelihood of evaporation within the pump. The engineer then tried different combinations of system pressures and temperatures, to ensure that the refrigerant at both the inlet and outlet of the pump was subcooled. The system's water circuit, which used a chiller to simulate a city water supply at constant temperature, was also modified to add a simple cooling jacket to the pump head, in hopes of reducing pump head temperatures.

Figure 40 illustrates the first successful test of the system with a steady refrigerant pump operation. The system pressure was kept low since this was an initial test, but other operating parameters were adjusted accordingly to operate the system components.

Table 6 summarizes the results of the test. Though this was a brief test during a time of day when insolation was decreasing, the test demonstrated:

- Successful operation of solar collector, glycol pump, refrigerant pump, generator, and condenser as a complete system.
- Ability of the generator to heat, boil, and superheat R-134a.
- Ability of the condenser to cool, condense, and subcool R-134a.
- Heat balance in both the generator and condenser, which demonstrates the function and accuracy of the chosen instrumentation (for example, water and refrigerant flow meters).

Figure 40: Solar Subcomponent Initial Test



Source: Altex Technologies Corp.

Table 6: Solar Subcomponent Heat Balance Results

	Collector		Generator				Condenser			
	Glycol In	Glycol Out	Glycol In	Glycol Out	R134a In	R134a Out	R134a In	R134a Out	Water In	Water Out
Temp (^{O}F)	147.2	159.0	155.2	149.4	81.9	144.0	107.0	67.4	46.8	47.3
Refrigerant SH / SC	-	-	-	-	44.0 (SC)	18.3 (SH)	32.8 (SH)	6.6 (SC)	-	-
Pressure (psia)	-	-	63.0	62.5	201.8	201.4	-	-	-	-
Enthalpy (Btu/lb)	67.4	72.8	71.3	68.4	97.7	184.9	184.9	97.7	14.9	15.5
Heat Transfer (Btu/hr)	2359.3		1168.8		1083.1		1088.6		1096.1	

Source: Altex Technologies Corp.

As shown the energy added to the collector and the energy added to the refrigerant were substantially different. The glycol lost half of the added energy due to heat loss in the piping from the roof-mounted collector to the lab system. The piping length was more than twenty feet, and was not yet insulated in this preliminary test. In a field installation of BEECH, the collectors and the remainder of the system would be located much closer, and the piping would be insulated. In all subsequent solar testing, these lines were insulated.

After this initial successful test, pump operation became more sporadic—it would prime and operate briefly, but reliable operation could not be achieved. Finally, the pump was removed from the system and tested per the manufacturer's recommendations, and determined to be defective. The manufacturer serviced and returned the pump, and Altex staff reinstalled the pump.

While the pump was being serviced, Altex technicians made a number of changes to the system, hoping to eliminate all potential causes of non-priming. They replaced the vertical refrigerant reservoir with a horizontal unit, and relocated it below the condenser outlet. Additional sight glasses were added up- and downstream of the pump to monitor for vapor bubbles. The pressure regulator was also replaced with a needle valve, as noted above, and when the pump was returned, it was reinstalled approximately three feet lower than the reservoir, to increase the pressure head on the pump inlet.

When the rebuilt system was tested, priming seemed marginally improved, but was still unreliable. Intermittent operation could be achieved by varying the system refrigerant charge pressure and cycling the pump, but continuous operation was not achieved. Finally, the test engineer again removed the pump and found it to be non-functional, in the same manner as was supposedly addressed by the pump manufacturer's warranty service.

At this time, the subcomponent testing was behind schedule, so the system was reconfigured without the pump to create a test apparatus that could measure solar collector performance at the critical, high-temperature operating points required for maximum BEECH efficiency. The chilled water supply was used instead of refrigerant to absorb heat from the solar loop. This allowed accurate control of the refrigerant temperature at the inlet to the solar collector.

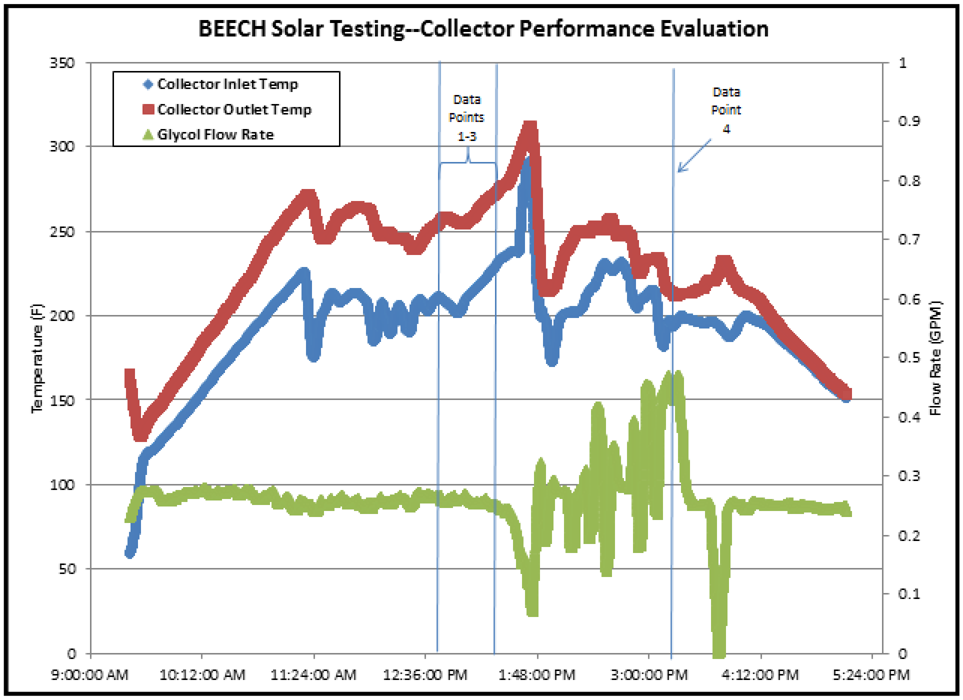
Figure 41 shows a sample test from this configuration, and Table 7 summarizes the operating conditions at selected points, using five minute averages of the data.

The system operated with a steady glycol flow rate throughout the morning, and peak temperatures were reached around 1:30 pm. At this time, system temperatures had increased above the test limit of 300˚F (149˚C). The sharp change in glycol temperature at this temperature indicates highly variable conditions, likely due to localized boiling. The test engineer then adjusted glycol flow via the manual valves in the system, attempting to reach a new steady-state point at a higher low rate. However, steady state conditions were not achieved again, partially due to the decreasing sunlight in the winter afternoon.

In a commercial or demonstration system, the manual valves would be replaced by automated units, thus allowing smooth adjustment of the glycol flow throughout the day and also avoidance of boiling. The critical process parameters for predicting full system performance based on collector performance are depicted, and the test points at which they were recorded are noted. Test Point 2 is of interest, as it reflects the highest energy input of the four points;

test point 3 is also of interest, as it reflects the highest collector outlet temperature. Higher temperatures were achieved, but during highly-transient operation.

Figure 41: Solar Collector Performance Evaluation



Source: Altex Technologies Corp.

Table 7: Solar Collector Test Results

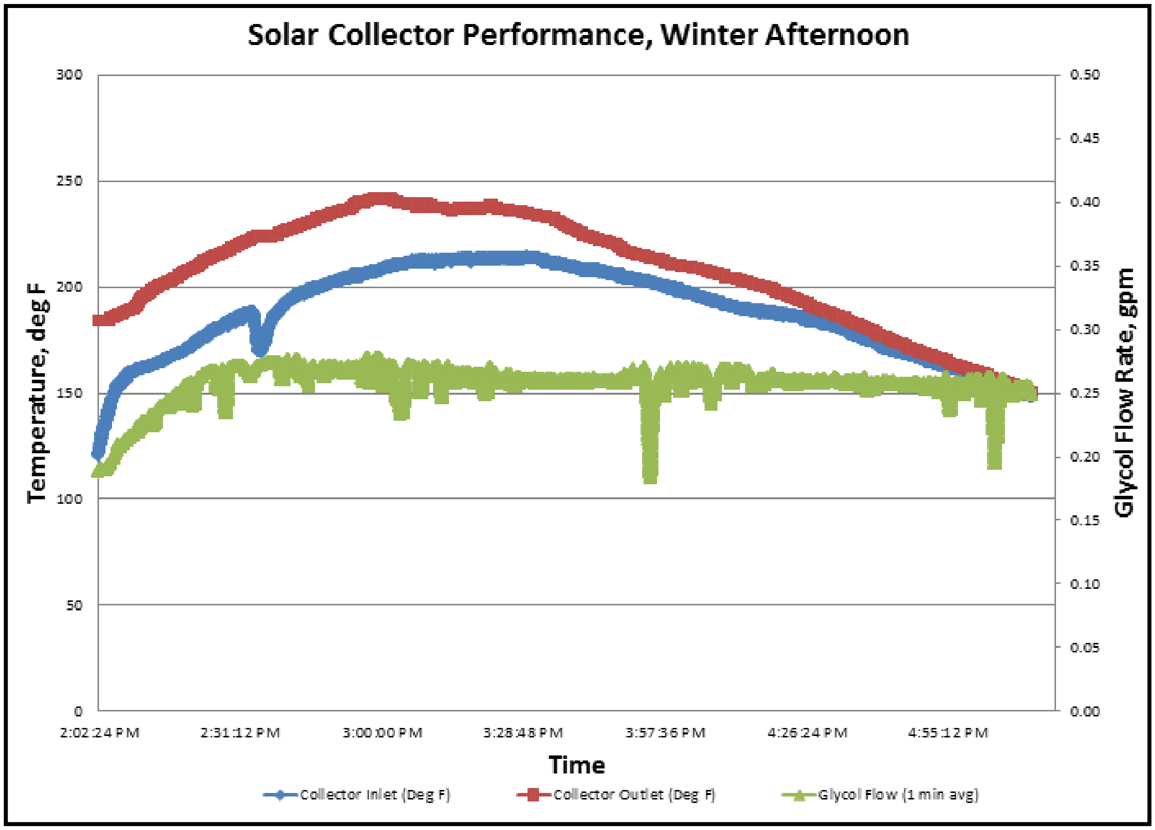
Test Point	Flow Rate *GPM*	Solar Collector In ^{o}F	Solar Collector Out ^{o}F	Q *BTU/hr*
1	0.27	211.4	252.0	**5316**
2	0.26	201.7	255.6	**6330**
3	0.25	228.2	272.4	**5069**
4	0.43	194.9	213.9	**3730**

Note: Test point 4 recorded @ 3:15 pm, well after peak insolation.

Source: Altex Technologies Corp.

Since the previous testing did not accurately capture the full day's operation, the test engineer retested the system at a similar glycol flowrate (Figure 42). Peak collector temperatures were lower, but the boiling issue was avoided. Minimal engineer intervention was required during this test, and the collector, glycol pump, and the flowmeters all performed as expected.

Figure 42: Solar Collector Test—Steady State Glycol Flow

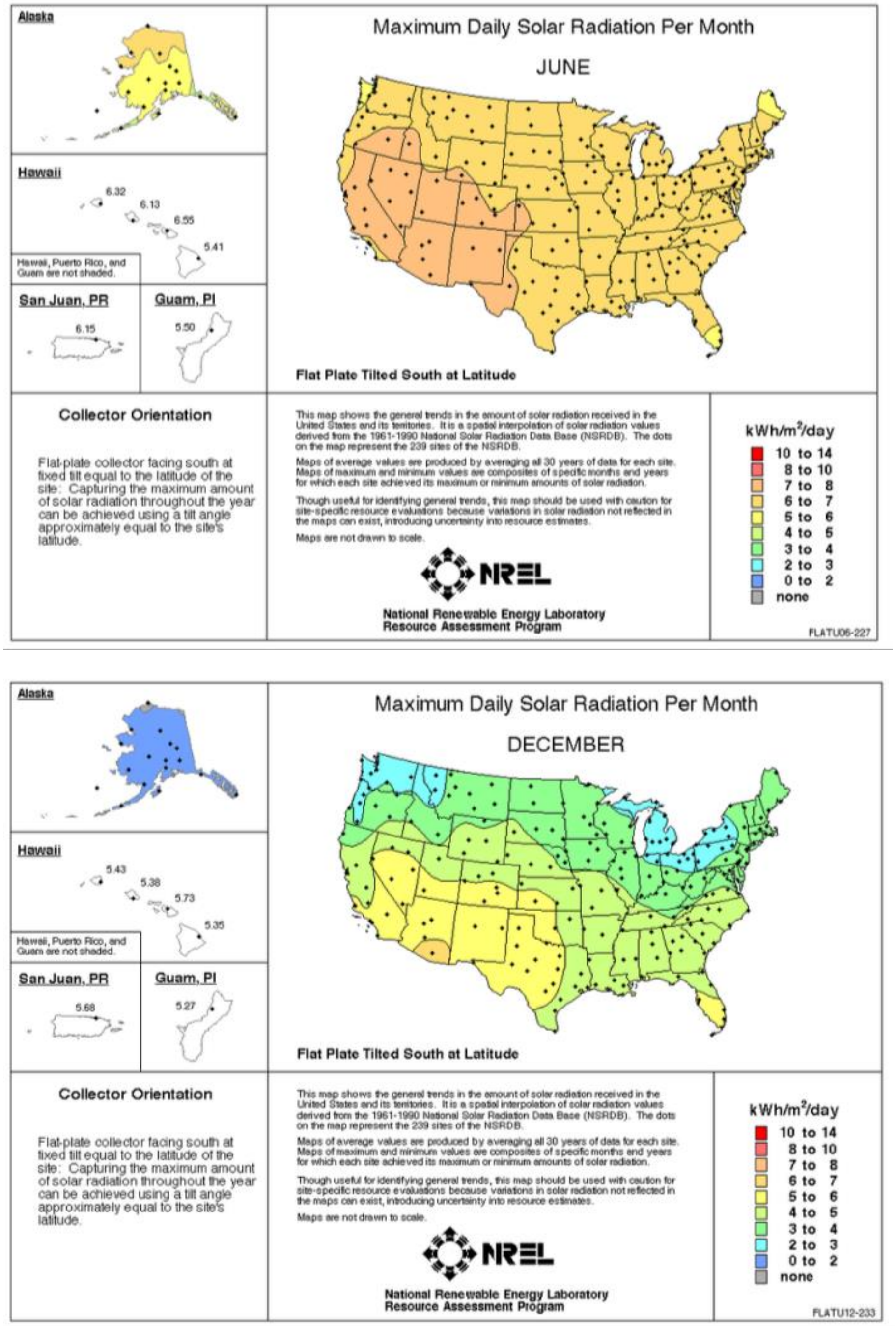


Source: Altex Technologies Corp.

The data illustrate the difficulty in measuring system performance while thermal input is changing. The steady-state operating conditions that are required to accurately calculate heat balances and heat exchanger performance cannot be achieved while the sun is constantly moving. Therefore, five minute averages of data are used in the calculations presented in Table 7, as noted above. In a more mature system with electronically-controlled flowrates, more consistent operation can be expected. However, the data gathered here were adequate for analysis and for input to the CHEMCAD process model for predicting full-system performance under similar conditions.

As expected, the collector did not achieve the maximum 10,000 Btu/hr performance predicted by the manufacturer for summer operation. The peak measurement was 6,330 Btu/hr. Higher performance could likely have been achieved with process tuning, but a more meaningful factor is that the testing was performed in December and January, when less solar energy is available. Ambient temperatures are lower, leading to greater heat losses at the collector, and a collector that is angled for maximum performance during summer months (when cooling demand is highest) is expected to have lower performance in winter months. As an additional reference, Figure 43 presents 30-year averages of solar radiation, as reported by the National Renewable Energy Laboratory.

Figure 43: National Renewable Energy Laboratory Solar Insolation Maps, Monthly Maximum for Fixed-Plate Collectors



Source: National Renewable Energy Laboratory, http://rredc.nrel.gov/solar/old_data/nsrdb/1961-1990/redbook/atlas/serve.cgi

Maximum radiation in California during June (7-8 kWh/m2/day) is substantially higher than during December (4-5 kWh/m2/day). While this averaged historical data cannot be used to predict performance of a collector on any particular day, especially one with varying cloud cover, it does illustrate the seasonal averages. Since cooling demand is driven by ambient temperatures and solar radiation (i.e. more cooling is needed in summer), it is acceptable for

BEECH to produce less cooling in winter. In fact, BEECH can be operated to produce only hot water if there is no demand for cooling, thus maintaining a useful output to offset the facility's overall energy consumption.

In summary, the solar collector testing demonstrated that:

- Commercially-available evacuated tube collectors can achieve and operate at the high temperatures required for BEECH
- Standard balance of plant hardware (expansion tanks, pump station, and piping) can be used, and were capable of day-long operation with no noted issues
- Collector outlet temperatures can be kept at acceptable levels by manually adjusting refrigerant flow rates between 0.25 gpm and 0.43 gpm per panel, which is a suitable function for a future automatic control system.

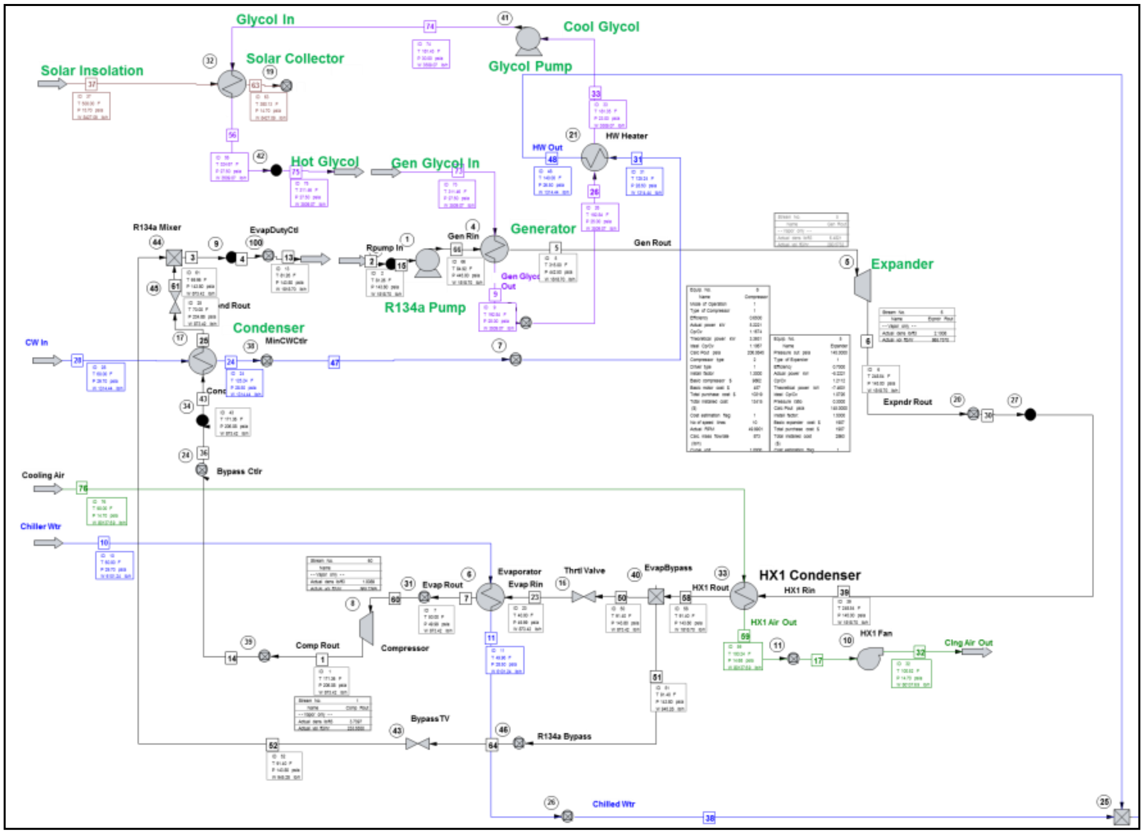
Prediction of Full-Scale Solar System Performance

As part of Task 2 system design activities, Altex engineers created a complete BEECH process model using CHEMCAD software. That model, in conjunction with the water demand study included in the Site Specification Report, predicted that a 30-collector array would be appropriate for producing 5.0 refrigeration tons of cooling (60,000 Btu/hr) and 3.7 gallons/minute of hot water. A schematic of the process model is shown in Figure 44, and the solar system components tested or simulated in this Task have been highlighted with green text.

These two points of high interest provided collector performance at two different collector outlet temperatures, and at approximately the same glycol flow rate. Altex engineers first verified that the CHEMCAD fluid model for the glycol mix matched the properties of Tyfocor LS, the specific brand provided by Kingspan for this test. Then, they entered the experimental temperatures into the model and ran all unit operations. Desired expander and compressor pressures were kept constant for both analyses. The model was constructed based on the 5.0 refrigeration tons/3.7 gm hot water target. Since the solar testing was based on a single-collector test, the experimental flow rate of glycol per panel could be used to scale the cooling and heating outputs based on the CHEMCAD-predicted glycol flow rates necessary to achieve those outputs, under these temperature conditions. A full, thirty collector system was used as the basis for the scaling. The results of this modeling and scaling are shown in Table 8.

As expected, Point 2, which had a higher thermal input, was capable of producing a higher cooling and thermal output. The higher outlet temperatures of Point 3 would permit a slightly smaller generator heat exchanger, but in a practical system, the heat exchanger would be specified to be slightly oversized, to accommodate a range of conditions. The system target of 5.0 tons/3.7 gpm was not achieved with a thirty collector system, but the 2.9-3.6 tons/1.6-2.0 gpm outputs are still recovering approximately 2/3 of the input energy. As noted, the data was recorded in December and January, and the manually-operated system was not fully optimized. Summertime operation with a mature electronic control system would increase both outputs.

Figure 44: CHEMCAD Solar System Process Simulation Schematic



Source: Altex Technologies Corp.

Table 8: Solar System Model Predictions

	Test Point	2	3
Test Data	Glycol Flow (gpm)	0.26	0.25
	Collector In (^{O}F)	201.7	228.2
	Collector Out (^{O}F)	255.6	272.4
	Solar Energy Added (Btu/hr)	6330	5069
Full System	Glycol Flow, 30 Panel Array (gpm)	7.77	7.56
	Solar Energy Added (Btu/hr)	189,900	152,070
CHEMCAD Simulation, Full System	Cooling Output (tons)	3.64	2.90
	Hot Water Output (gpm @ 140^{O}F)	2.00	1.59
	Cooling Output (btu/hr)	43,693	34,785
	Hot Water Output (btu/hr)	80,103	63,773

Source: Altex Technologies Corp.

Conclusions: Solar and Subcomponent Testing

Though pump issues did not permit extended operation of the subcomponent test system, the functional testing of other key components and test equipment was successful. The testing also provided adequate outputs to evaluate full system solar performance, based on the CHEMCAD model. In summary, the testing demonstrated:

- Successful operation of solar collector, glycol pump, refrigerant pump, generator, and condenser as a complete system, for short time intervals.
- Ability of the generator to heat, boil, and superheat R-134a.
- Ability of the condenser to cool, condense, and subcool R-134a.
- Heat balance in both the generator and condenser, which demonstrates the function and accuracy of the chosen instrumentation (for example, water and refrigerant flowmeters).
- Commercially-available evacuated tube collectors could achieve and operate at the high temperatures required for BEECH.
- Standard balance of plant hardware (expansion tanks, pump station, and piping) could be used, and were capable of day-long operation with no noted issues.
- Collector outlet temperatures can be kept at acceptable levels by manually adjusting refrigerant flow rates between 0.25 gpm and 0.43 gpm per panel, which is a suitable function for a future automatic control system.
- The conditions achieved in the test could, in a full 30-panel system, produce up to 3.7 tons of cooling and 2.0 gpm of hot water.
- The Max Machinery flow meters operated well throughout testing, and would be re-used in full system testing.
- The full-scale system required an alternative refrigerant pump, either of a different type altogether, or a diaphragm pump from a different manufacturer.

CHAPTER 9: Waste Heat System Assembly

Initial Assembly

In addition to the components noted in the previous sections, the Altex team assembled other supporting parts. Some parts were modified to add instrumentation ports or sight glasses, such as the oil reservoir and refrigerant receiver tanks shown in Figure 45.

Figure 45: Oil and Refrigerant Tanks, Modified with Sight Glasses

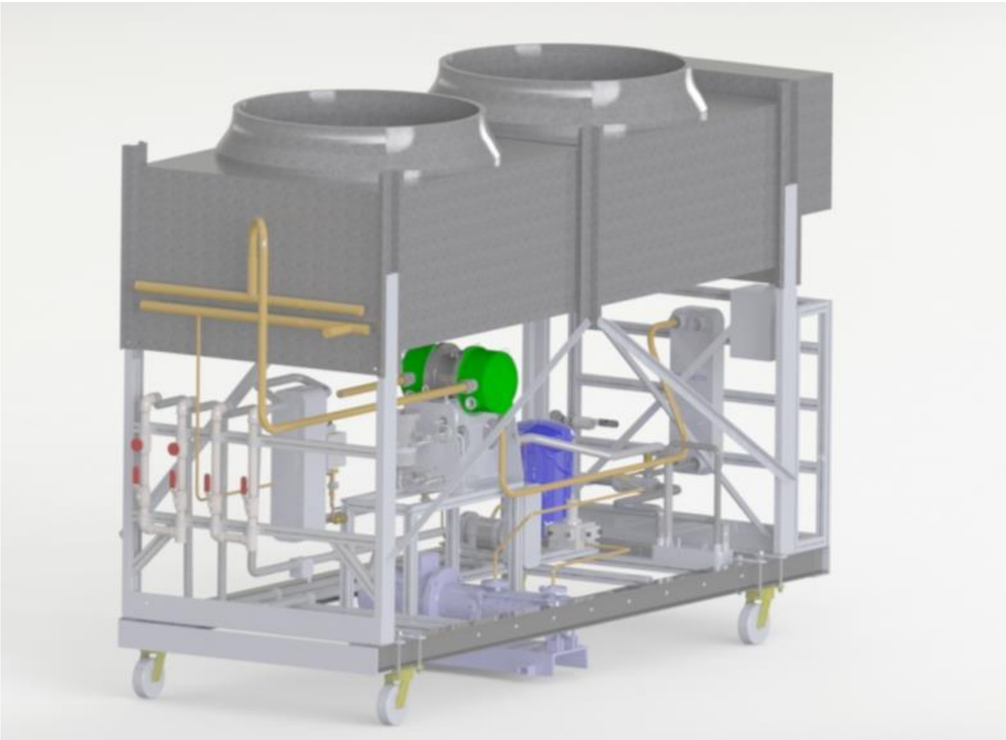
Source: Altex Technologies Corp.

Minor subcontractor Legacy Chiller provided assistance in procuring the brazed plate heat exchangers, as well as the air-cooled condenser shown in Figure 46. The condenser and its frame is the largest component of the BEECH system, and it was modified to be the framework and support structure for all of the system components outside of the heat recovery system. Altex engineers created a CAD model of the system, as shown in Figure 47 with all major components and piping, and reviewed this layout with Legacy Chillers staff, who provided feedback based on their experience building and selling commercial chiller systems.

Figure 46: Refrigerant Condenser, Received from Legacy Chillers

Source: Altex Technologies Corp.

Figure 47: Solidworks Design of the BEECH System



Source: Altex Technologies Corp.

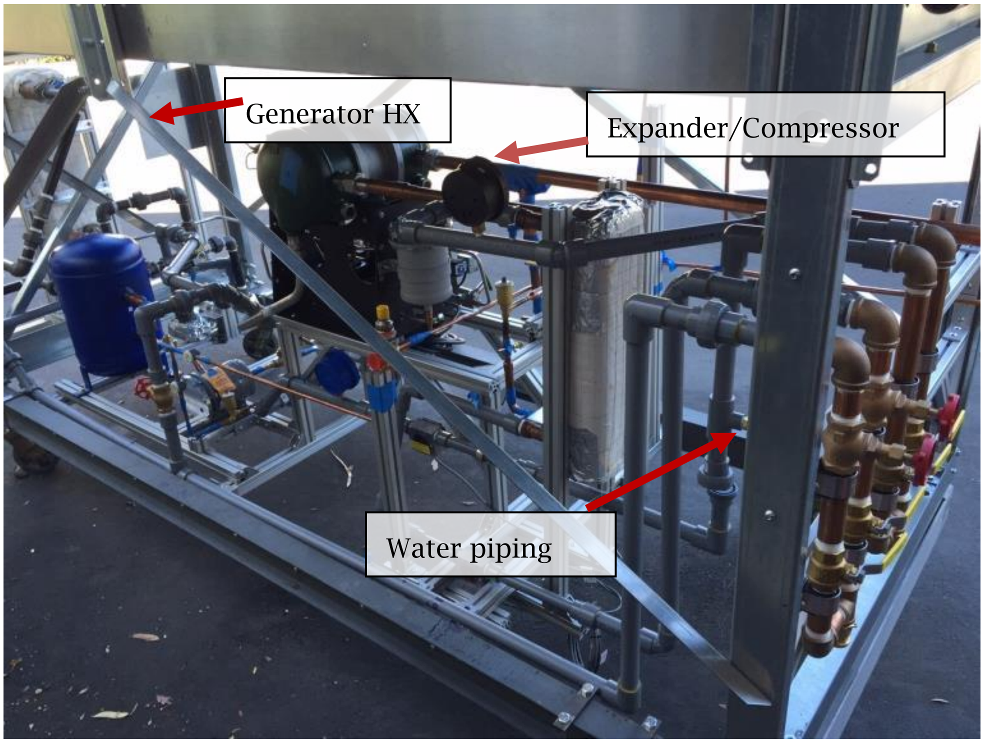
Assembly began with fabrication and installation of an aluminum sub-frame to support the various components. The assembled frame, with two heat exchangers and the refrigerant tank already installed, is shown in Figure 48. After all major components were placed, piping and tubing were installed, fit checked, and then glued, soldered, or brazed in place, as appropriate. Figure 49 shows the system during this phase, with the gray plastic PVC lines completed for the water system, and the copper lines ready for brazing. The system was then insulated, using fiberglass, closed-cell EPDM, and Microtherm insulation where appropriate. Simultaneously, wiring routes were established, and instrumentation was installed in accordance with the data acquisition plan. Figure 50 shows the electronics box, which contains the system power distribution, the National Instruments cards, and various valve controllers and controls devices. The system was controlled with National Instruments Labview software and hardware. The same equipment was also used for test data was acquisition Altex engineers created a custom program and user interface, as shown in Figure 51.

After system assembly was completed, the Altex team performed system validation and functional testing, as summarized in Appendix A.

Figure 48: System Frame Assembly

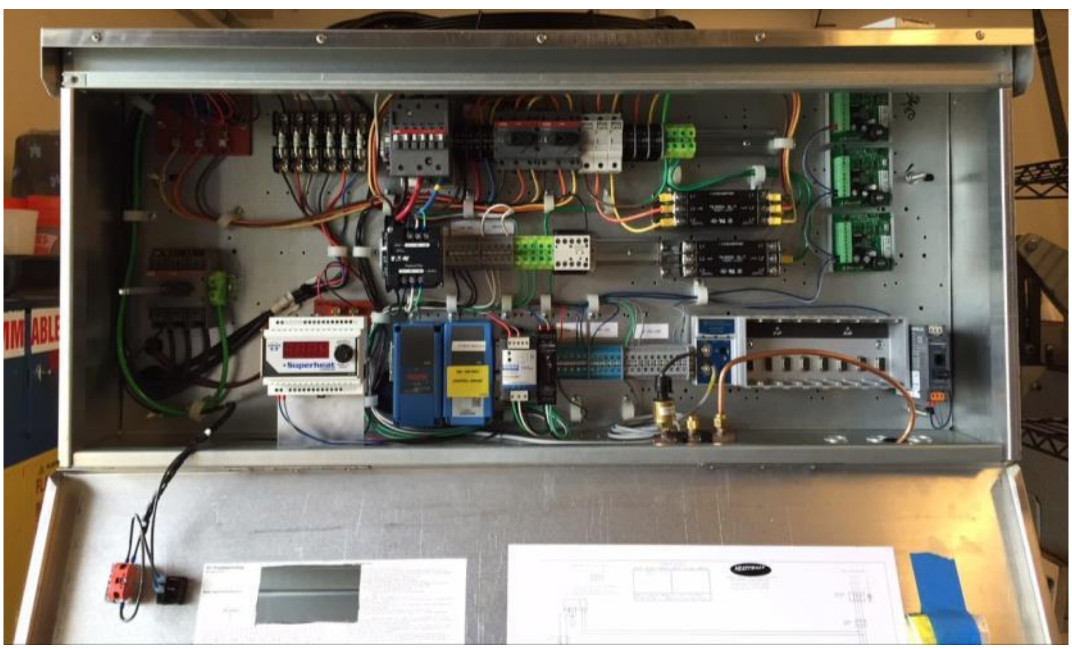
Source: Altex Technologies Corp.

Figure 49: System During Piping Installation



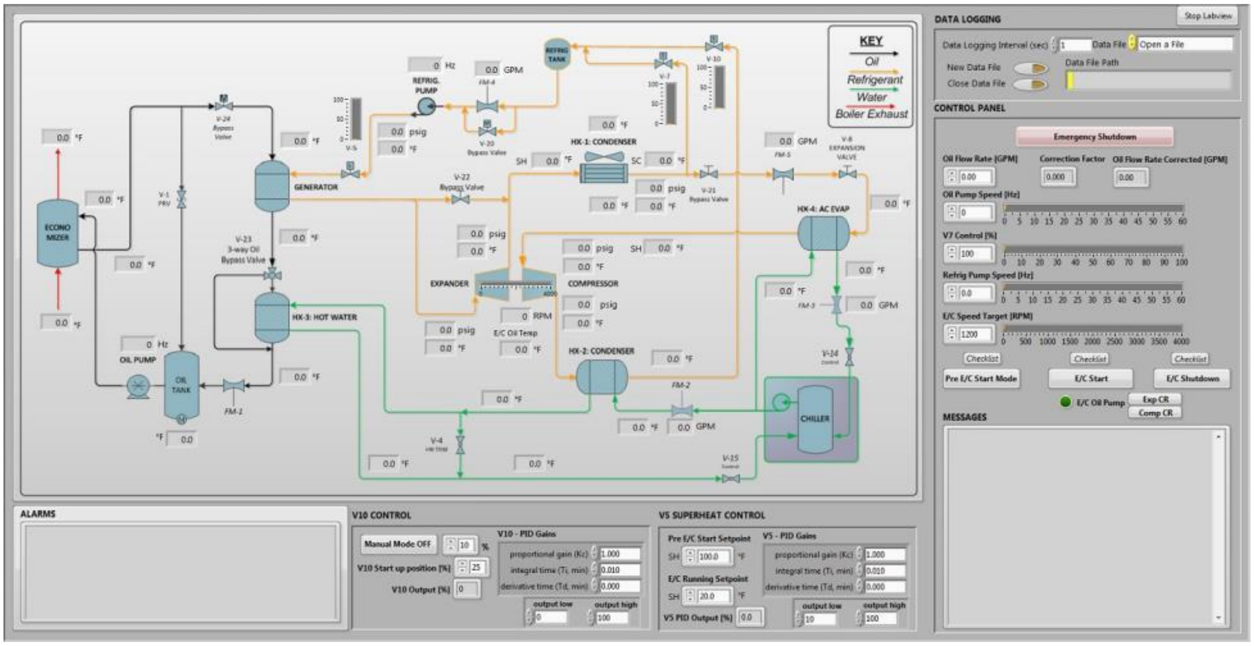
Source: Altex Technologies Corp.

Figure 50: System Electronics Box during Assembly



Source: Altex Technologies Corp.

Figure 51: Labview User Interface



Source: Altex Technologies Corp.

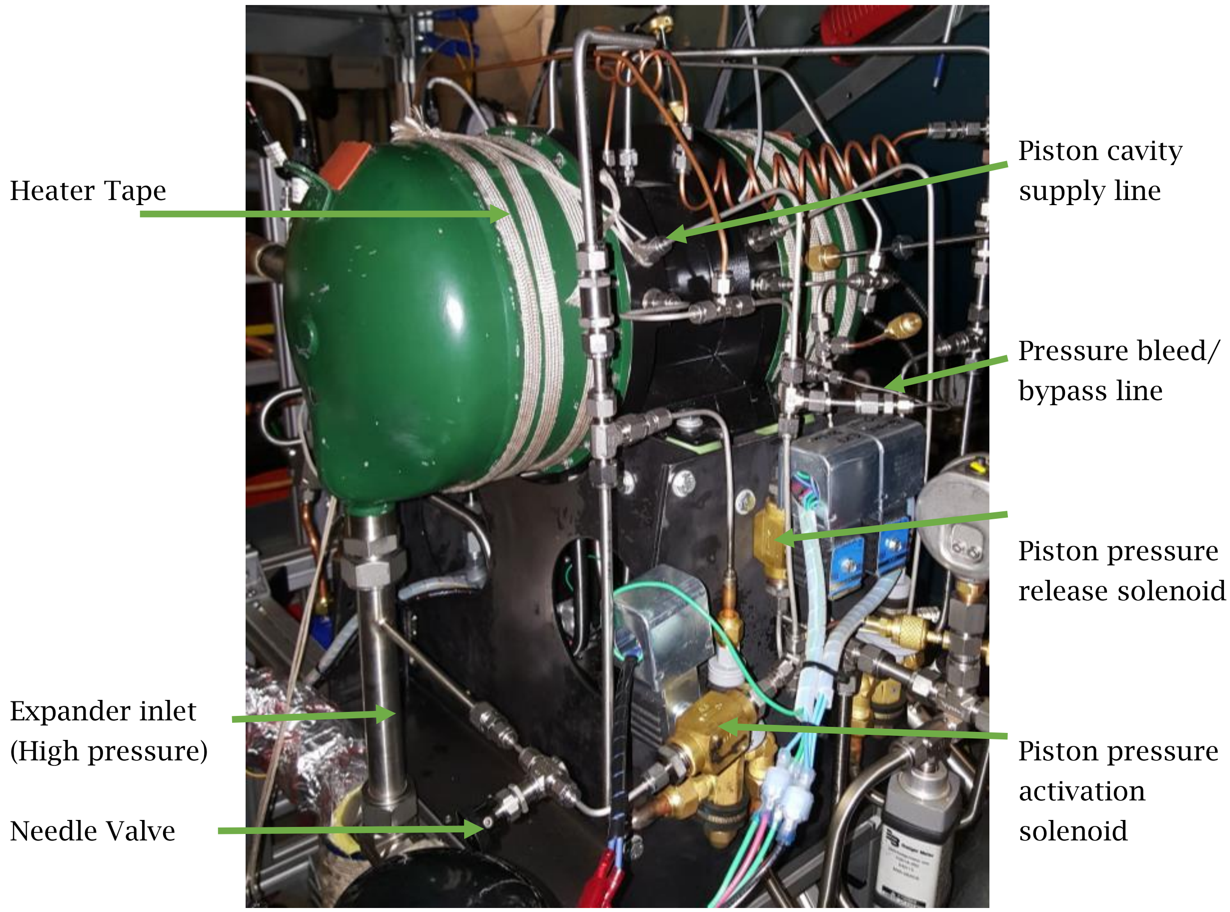
System Mechanical Refinement

Upon successful completion of verification and function testing, the test program proceeded to complete system functional testing. As testing progressed, it became obvious that expander/compressor start-up was a major challenge, and this problem had to be resolved before any system optimization could be pursued. This section focuses on mechanical refinements made to the system throughout the course of testing, with brief descriptions of the test activities provided for context. The following chapter will discuss test results in further detail.

Initial system fill with refrigerant indicated liquid refrigerant migration issues (prior to system warm-up), and these were solved using heater tapes located on the expander compressor tank heads and oil reservoirs, as shown in Figure 52.

Initial system start-up attempts were unsuccessful, even though adequate refrigerant flow, pressure, and temperature were achieved at the expander inlet. Refrigerant pumping and pressurization, vaporization, and condensation were all successful, with the system operating in a start-up “bypass mode.” In this mode, the pressurized refrigerant vapor is routed from the generator outlet to the condenser inlet via a pressure letdown valve, rather than flowing through the expander. The planned operating sequence was to start in bypass mode, achieve and verify proper refrigerant flow, pressure and temperature, and then close the bypass valve and route the refrigerant to the expander, thus starting expander operation. However, when this was attempted, the expander did not rotate.

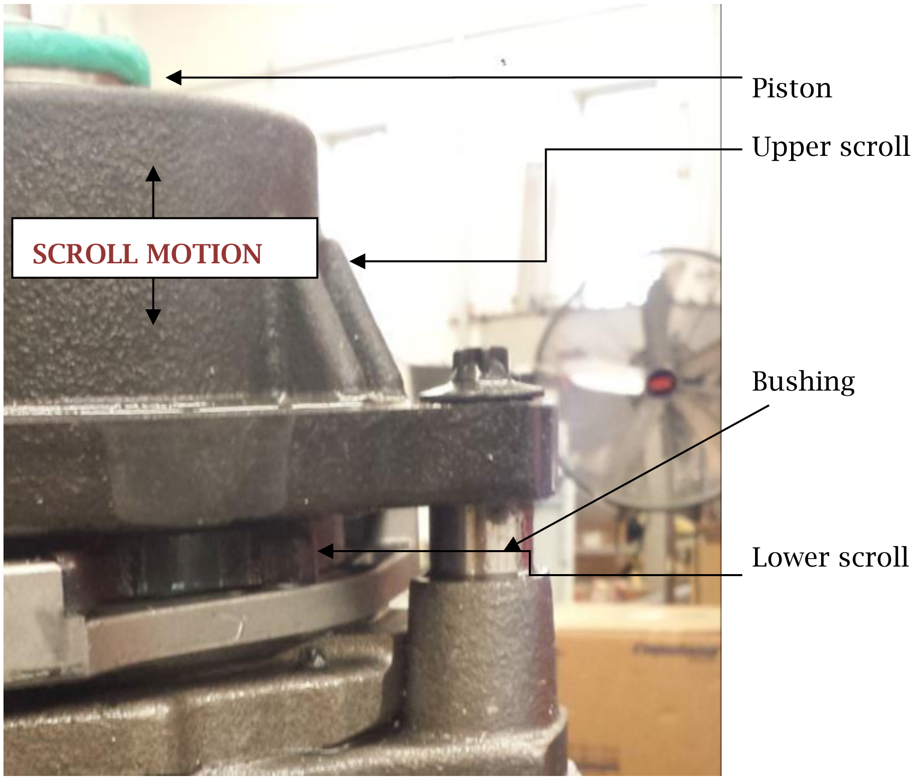
Figure 52: BEECH Expander/Compressor, as Modified for Testing



Source: Altex Technologies Corp.

To perform root cause analysis of the poor startup characteristics of the expander, Altex engineers disassembled the expander/compressor for inspection. They determined that the upper scroll was binding on the bushings, shown in Figure 53. By design, the upper scroll is allowed to move axially. The piston shown at the top contacts the inside of the vessel, and a pressurized volume of refrigerant between the piston and the upper scroll forces the upper scroll against the lower scroll, and maintains an adequate seal. Since the BEECH expander/compressor is mounted in a horizontal position (in contrast to the vertical orientation of the standard refrigeration compressor from which this unit is derived), the weight of the upper scroll caused misalignment and binding.

Figure 53: BEECH Expander/Compressor, Scroll Motion Illustration

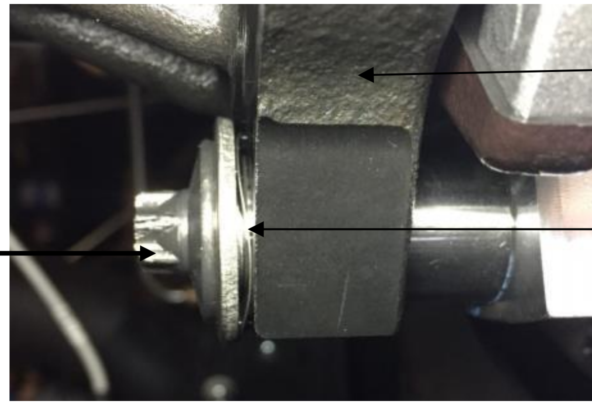


Source: Altex Technologies Corp.

To solve this problem, Altex engineers added wave springs between the upper scroll and mounting bolts (Figure 54). The two upper wave washers were shimmed to apply slightly greater force to the upper scroll, which counters the misalignment and binding due to gravity. The wave spring force was also adjusted to prevent the lower scroll from dropping and losing its seal when the expander/compressor stopped between the 8 o'clock and 2 o'clock orientations.

In a typical compressor application, the pressure in the cavity under the piston affects efficiency and wear. The piston cavity is pressurized by a port connected to a scroll cavity located in the middle of the compression cycle. The compressor manufacturer strategically locates this hole to balance efficiency and wear. In an expander application, the piston pressure not only affects efficiency at steady state running conditions, but also affects start up capability. If the pressure is too high, the scrolls will not move due to high friction. If the pressure is too low, the scrolls can separate from each other, which results in high internal leakage and poor starting characteristics. Altex engineers decided to design a system that allows full control of piston pressure, to maximize the chances of successful startup and with minimal impact on operating efficiency. This modification has been performed by other researchers who have modified scroll compressors for use as expanders.

Figure 54: Expander/Compressor Wave Springs Added to Solve Binding Issue



Source: Altex Technologies Corp.

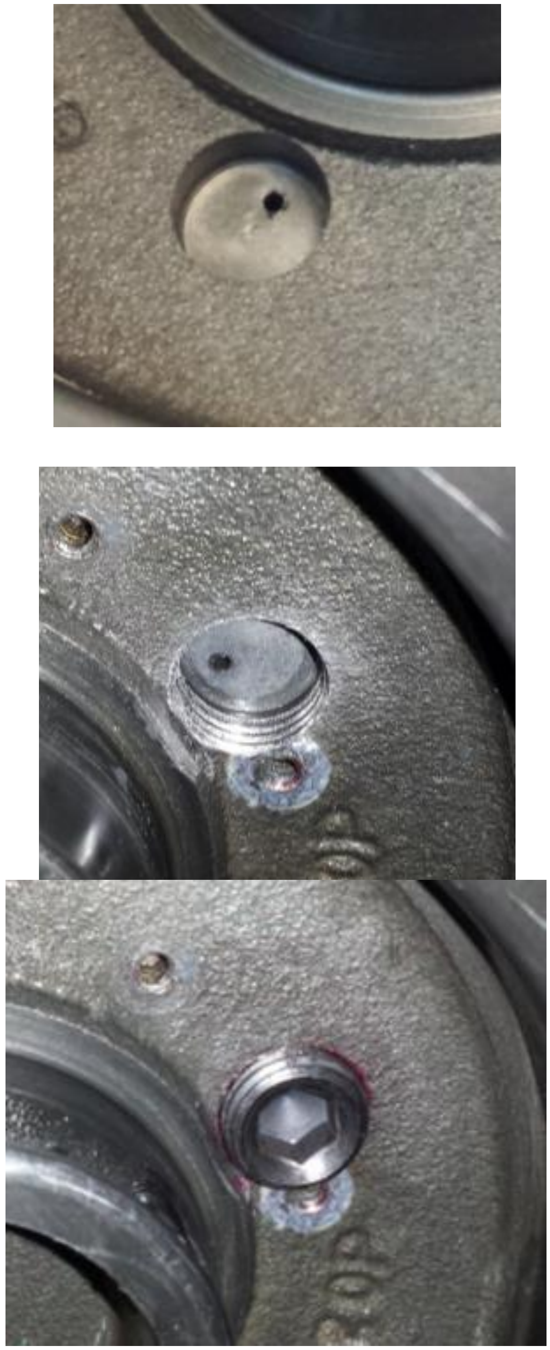
As shown in Figure 55, the upper scroll was machined to accept a plug, and that flow path was replaced by external tubing, with a manual needle valve for pressure control and solenoid valves for application or release of the pressure. The test engineer also updated the Labview control system to control the solenoid valves.

Even after these changes, the expander starting issue persisted. No rotation of the expander was detected, and repeated teardowns of the system (which required evacuating all refrigerant, performing mechanical work, then vacuuming and refilling the system for testing) became time consuming. A nitrogen test apparatus was then built, as described in Chapter 3. This device permitted rapid iteration of various operating sequences, piston pressures, and other parameters to assist startup.

During the nitrogen testing, an additional modification to the upper scroll was tested. As shown in Figure 56, a small (<0.060" diameter) hole was drilled from the piston pressure cavity into the cavities formed by the mating scrolls. This was intended to provide additional motive force to the scrolls at start-up, when mass flow is otherwise low and hydrodynamic sealing is less than would be experienced at full speed. This internal bypass, if successful, could be controlled via an external solenoid valve to operate only at start-up, thus eliminating any efficiency detriment during full operation. Testing on nitrogen showed no additional benefit, and so the port was plugged during all refrigerant-based testing. During one of the post-test inspections, Altex engineers noticed that after many tests that did not result in substantial flow of R-134a through the expander or compressor, the scrolls were dry.

As part of the assembly process, all scrolls are lubricated with refrigeration-grade oil. Under normal operation, additional oil is carried through the unit by the refrigerant, thus keeping the metal pieces lubricated, and limiting leakage between the scrolls. To verify the potential effects of low lubrication on start-up, engineers attempted one start-up on nitrogen using the dry scrolls. At this point in the test program, conditions for reliable start-up on nitrogen had been identified, and these were used in the "dry scroll" test. Testing showed that start-up was more

Figure 55: Piston Pressure Port Modifications



Stock hole (top), Modified for Plug (middle), Finished Modification (bottom).

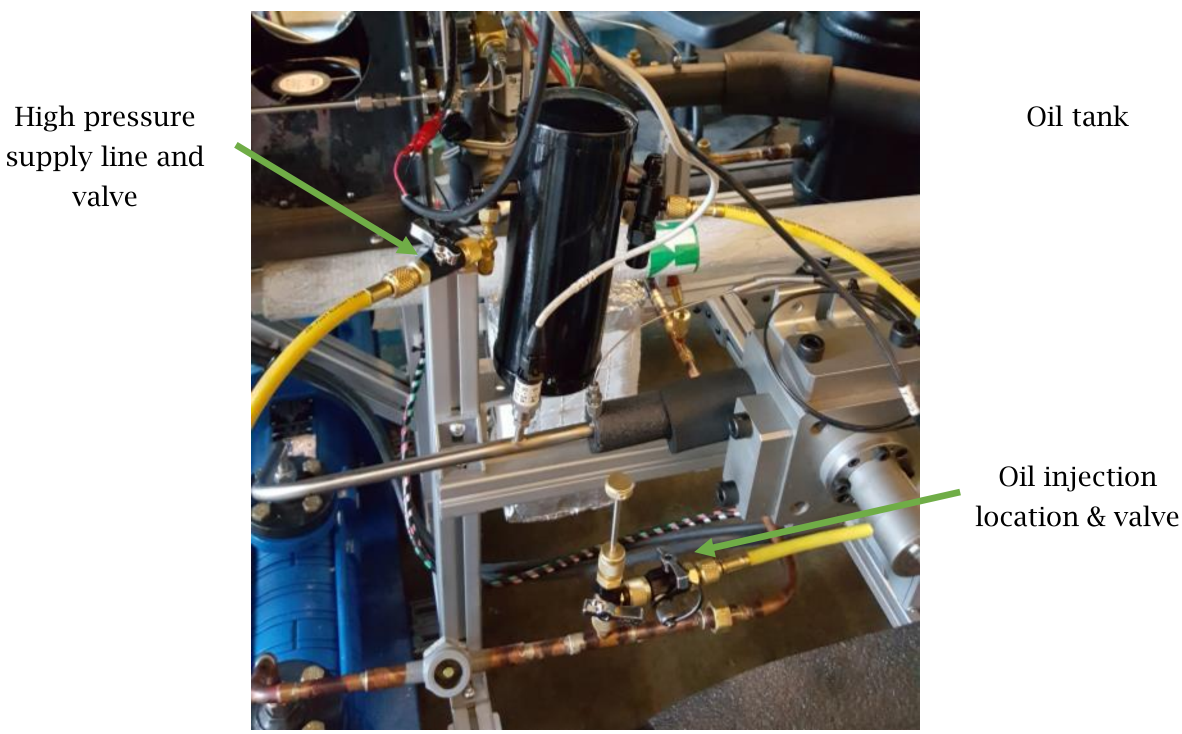
Source: Altex Technologies Corp.

difficult, though still possible. Subjectively, the expander/compressor acceleration and maximum speed noticeably decreased. This was not believed to be the root cause of the start-up issues, but still needed to be mitigated to reduce wear and eliminate any barrier to start-up during repeated testing. Therefore, Altex engineers designed and installed an oil injection system upstream of the expander inlet, and used this system during subsequent tests (Figure 57).

Figure 56: Mid-scroll Pressurization Port

Source: Altex Technologies Corp

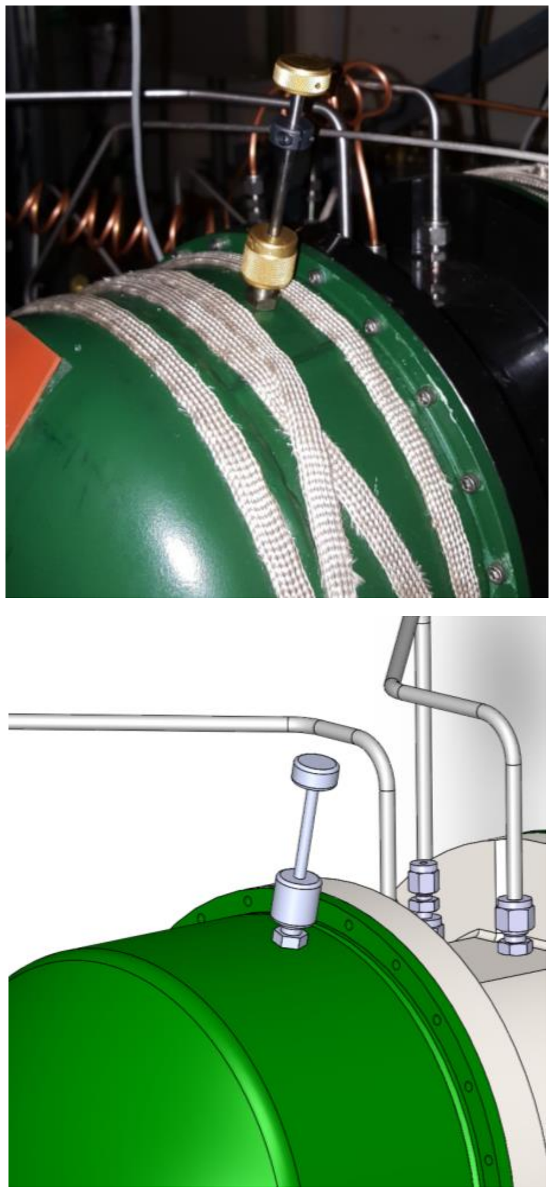
Figure 57: Manual Oiling System



Source: Altex Technologies Corp.

In another system modification, Altex engineers integrated a manual, “assisted start” system. After nitrogen testing had identified scroll positions and piston pressures that would reliably start the expander/compressor on nitrogen (and those positions where startup was not probable), engineers designed a mechanical assist that would initiate movement by pushing on the compressor-side lower scroll at four locations, thus enabling movement at all relative scroll orientations. The mechanical assist system had to be hermetically sealed, but still allow access from outside the vessel. As shown in Figure 58, four flare fittings were welded onto the compressor side tank head at strategic locations, and four shaft/O-ring assemblies were installed.

Figure 58: Mechanical Assist System—CAD design (l) and Implementation (r)



Source: Altex Technologies Corp.

The four shafts were fitted with stop collars to control the range of shaft movement. When the system is pressurized, a shaft can be pressed in by hand or lightly struck with a hammer to initiate movement of the scrolls; the shaft then self-returns to its starting position, due to the internal pressure in the vessel. Engineers tested the system using nitrogen and were able to start the expander/compressor from all angular positions. However, the results could not be duplicated with hot R-134a.

CHAPTER 10: System Testing

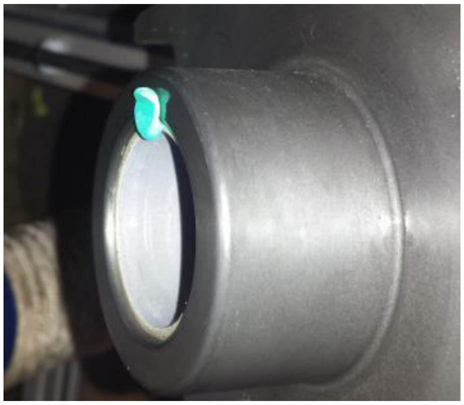
Altex engineers first performed multiple shake-down and subcomponents tests related to the boiler (waste heat source), condenser, and refrigerant pump. The firetube boiler in the test facility is rated for 10 x10^6 Btu/hr thermal input, and hot exhaust can be accessed at the ends of the first and second boiler passes. For maximum test flexibility, the exhaust heat recovery heat exchanger was placed at the end of the first pass, where the products of combustion (POC) are typically 1300-1800°F. This hot flow is then mixed with variable amounts of ambient air via a dilution blower and mixing manifold to create a range of exhaust volumetric flows and temperatures, based on the boiler burner's thermal input and the amount of dilution air. An additional blower was added to the dilution system to increase total dilution flow, to permit testing at the heat exchanger inlet temperatures of interest. Engineers also modified the wiring of the BEECH system's condenser fans to permit full control via the Labview control system. Before testing, engineers also implemented alarms in Labview to trigger audible and visual indicators in the lab to ensure safety and system integrity.

Expander/Compressor-Related Testing

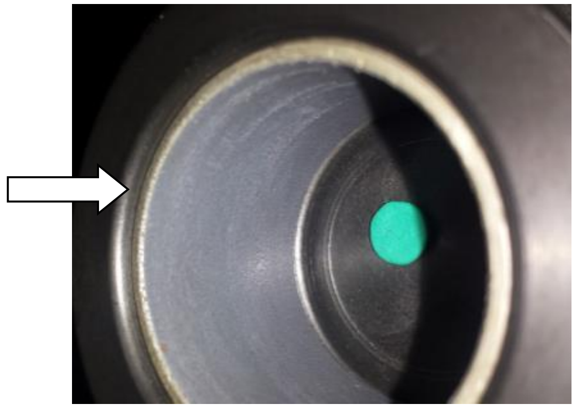
As noted in Chapter 9, the majority of system testing has concentrated on root cause analysis of the startup problem. The first round of analysis included the following actions:

- Verified superheat of refrigerant vapor (via measurement and observation of the sight-glass).
- Observed speed sensor indicator to confirm no change of state during attempted starts.
- Varied bypass valve opening sequence and rate during attempted starts.
- Varied expander inlet pressure during attempted starts.
- Increased internal spring pressure to counteract gravity effect on upper scroll.
- Reduced expander piston spring pressure to reduce turning resistance.
- Re-measured critical clearances to verify no binding or interference as-assembled (Figure 59).
- Held multiple conference calls with subcontractor Legacy Chillers, to review status and potential root causes.
- Reached out to academic and industry resources to seek additional technical support.
- Built a nitrogen pressure test rig to simulate refrigerant pressure, without the need to evacuate and recharge the system between mechanical changes.

Figure 59: Clearance Checks Performed Using Clay

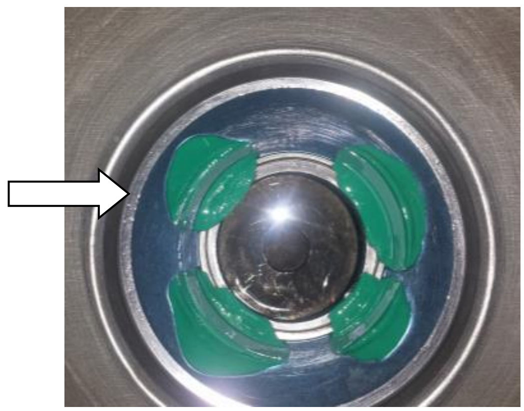


Clay placed under lower scroll



Compressed clay at interface of lower scroll and shaft

Clay placed in piston pressure chamber



Compressed clay at interface of piston pressure chamber and piston

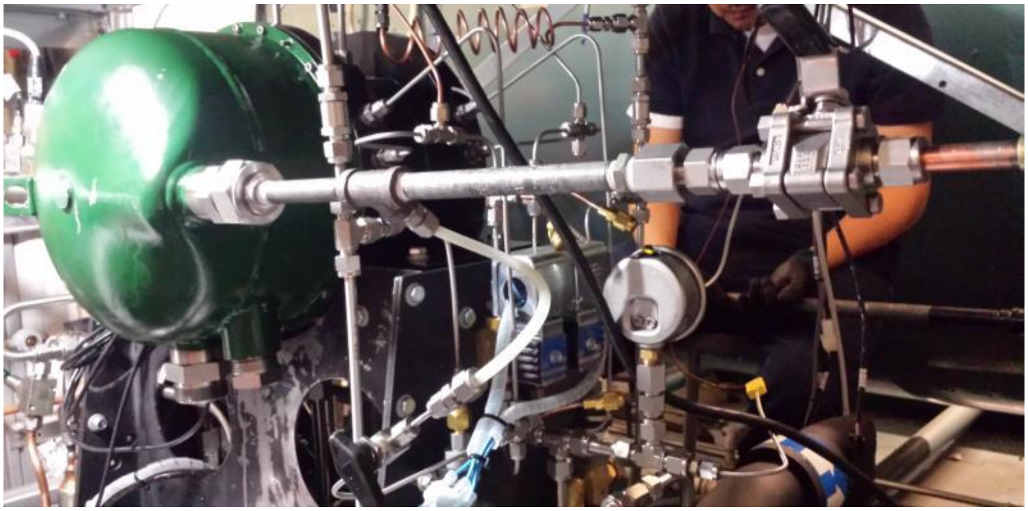
Source: Altex Technologies Corp.

The nitrogen test apparatus, as shown in Figure 60, was built to reduce the time required to iterate mechanical changes to, or inspections of, the system. For each change of a refrigerant-charged system, the R-134a must be recovered, the change implemented, and then the system must be sealed, vacuumed to remove air and water vapor, and re-charged with refrigerant. The complete process usually took two working days, depending on the complexity of the mechanical change.

Instead, a pressurized nitrogen bottle substituted for R-134a. Bottled nitrogen allowed engineers to test the expander/compressor in varying states of dis-assembly (and at various regulated inlet pressures), to evaluate the effects of mechanical or process changes without performing a full-system evacuation and refill with refrigerant.

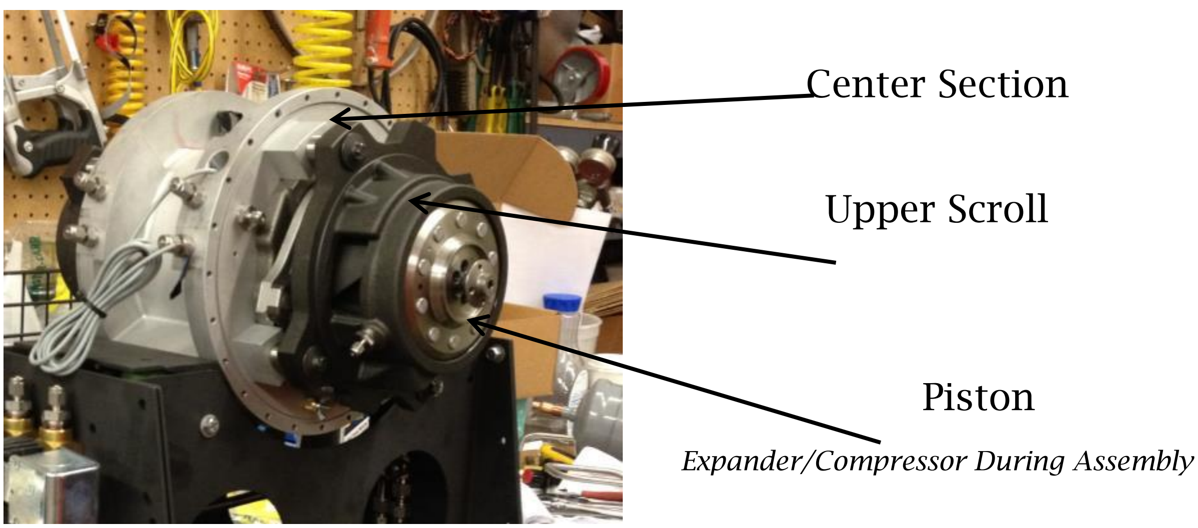
Based on published papers and input from subcontractor Legacy Chillers, piston pressure was the main focus of testing with nitrogen. As shown in Figure 61 and described in Chapter 2, the expander's upper scroll contains a piston that is actuated by refrigerant pressure and creates a seal against the inside of the vessel. The pressure of gas under the piston, and the rate at which the pressure is changed, is believed to be the major contributor to the starting issue.

Figure 60: Nitrogen Test Apparatus



Source: Altex Technologies Corp.

Figure 61: Expander/Compressor Component Illustration



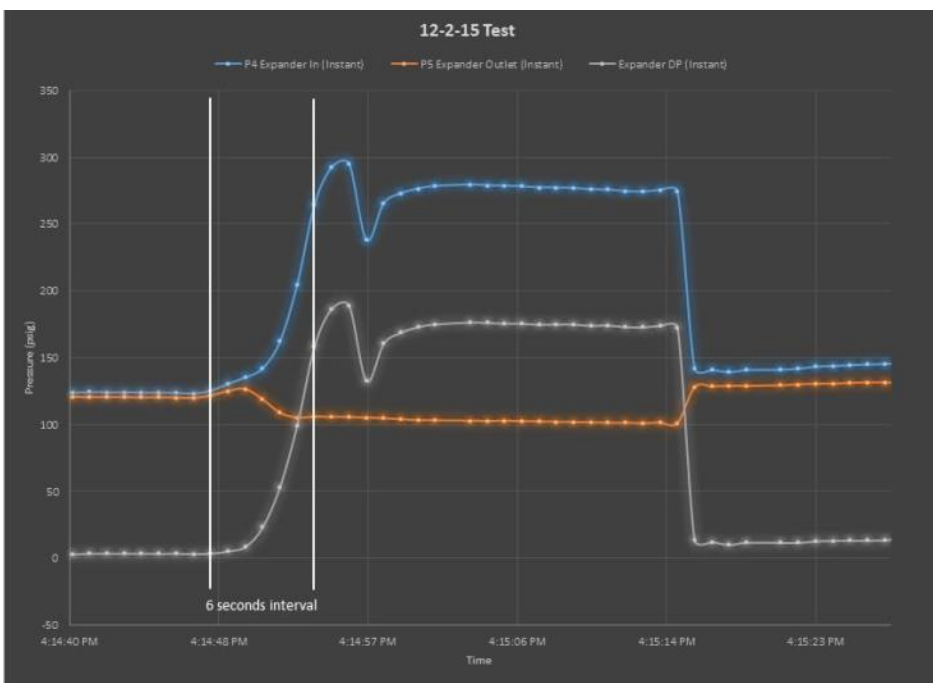
Source: Altex Technologies Corp.

Prior to startup, piston pressure is equalized to the expander outlet pressure through a 1/16" tube, and the majority of refrigerant flow is bypassed around the expander. When startup conditions (for refrigerant temperature and pressure) are reached, the test engineer closes the bypass circuit, thus directing all flow (or a greater proportion of flow) into the expander. Expander inlet pressure increases and outlet pressure decreases. However, experimentation showed that piston pressure reacted slowly, and lagged the expander outlet pressure rise. This results in high piston pressure (more than 30 psi) upon startup. Testing on nitrogen showed that a 30 psi piston pressure caused high static friction, which prevented the lower scroll from moving. At much lower piston pressures, the piston did not seal against the inside of the vessel, which allows refrigerant to bypass the expander and flow directly to the vessel outlet. Later testing determined that the optimum piston pressure was a range of 20-30 psi.

After a test program with nitrogen (summarized below) was completed, the lessons-learned were implemented in the refrigerant test plan. The system was then tested with refrigerant, and also with the compressor circuit pressure decreased by adding an expansion tank to the compressor outlet. Starting was not achieved. Data collected in this time period showed a six second elapsed time for the expander differential pressure (Inlet – Outlet) to increase to 160 psi. This pressure is within the operating window of the compressor unit from which the expander was derived. However, as shown in Figure 62, pressure increase in the first four seconds is only 50 psi. This indicates the potential for excessive friction, which will create excessive drag on the scroll and prevent or quickly stop movement.

In many of these later start attempts, the expander/compressor did rotate and that rotation was detected by the speed sensor installed in the unit. However, due to the sampling frequency of the data acquisition system and the response time of the sensor and its signal conditioning (converting the digital on/off sensor output into a voltage proportional to rotational speed), it was difficult to accurately determine the relative effect of the various changes based on rotational speed measurement alone. Instead, engineers monitored the speed sensor's built in light that illuminates when the counter weight was in close proximity.

Figure 62: Start-up Pressure Increase



Source: Altex Technologies Corp.

In summary, Altex engineers performed the following tests with pressurized nitrogen, and then with refrigerant:

- N_2: Measured piston pressure decay (baseline).
- N_2: Installed hardware (1/8" tubing that equalizes piston pressure to expander outlet pressure).
- N_2: Re-measured piston pressure decay.
- N_2: Varied expander inlet valve opening rate from instantaneous up to two seconds (open vessel, no tank head).
- N_2: Repeated expander inlet valve opening rate test (with tank head installed).
- N_2: Simulated mechanical –assist start by moving compressor side lower scroll.
- R-134a: Full system start attempt, with three different settings of the valve between the compressor outlet and the refrigerant tank.
- R-134a: Full system start attempt, with compressor-side pressure unloaded via external expansion vessel.
- Disassembled the system and noted that oil film was less than expected, which may have contributed to poor scroll-to-scroll sealing.
- N_2: Verified poor lubrication as a possible contributing cause to the most recent non-starts, by testing scrolls as-found, and then with typical lubrication. Starting ability and speed were shown to improve with proper lubrication.

Based on these results, the pressurized oiling system and the mechanical start-assist mechanism were implemented, as described in Chapter 9. After these changes, the team pursued the following test plan using R-134a:

- Quickly closed the bypass valve at or slightly before an assisted start attempt (this would increase expander differential pressure and force flow through expander).
- Started testing with a zero refrigerant pump speed and slowly increased speed while attempting to perform assisted starts, to determine if a lower-flow condition would assist starting.
- While the expander was in "max internal leakage mode" (believed to be leakage between the tank head and piston at low piston pressures), incrementally increased piston pressure while attempting assisted starts (to determine the pressure at which the piston sealed).
- Attempted starting with the compressor outlet valve open, to minimize compressor load.
- Activated compressor compression release solenoid (also intended to decrease compressor load).

- Opened valve at compressor inlet for a few tests, to create a small differential pressure across the compressor
- Injected oil upstream of the expander while attempting to start (for reduced friction)
- Operated oil pump briefly prior to start attempts (for reduced bearing friction)
- Kept oil pump on while attempting to start (for reduced bearing friction.)
- Continuously bump started and tried actively increasing piston pressure.
- Continuously bump started and tried gradually closing the bypass valve to build expander differential pressure.

The results and conclusions of that test program, and those previous, were:

- When piston pressure was too high, it would become more difficult to bump start and we would get partial rotation or in extreme cases none at all.
- At 120 psi expander differential pressure, starts were attempted at a range of piston pressures (0, 5, 10, 20, 30, 40, 50, 55 and 60 psi). Based on subjective evaluation by the test engineer, slightly more rotational movement was achieved as piston pressure increased to 20 – 30 psi. At 60 psi, it was obvious there was less movement and "self-start" potential. The conclusion was that there was not an obvious sweet spot, rather, there was a wide range of "lower" piston pressures that produce very similar results. If at any time bump starting was difficult, the test engineer reduced piston pressure to see if that was the cause.
- Starts were attempted at expander differential pressures of 65 to 250 psi. While the mechanical starting system could be used to start rotation at all pressures (and that rotation would be maintained beyond the range of motion imparted by the mechanism), the best results, as judged by duration of un-assisted rotation after initial assist, were achieved at differential pressures of 100-120 psi.
- During these tests, outlet pressure of the expander was maintained at or near 120 psig, and so the best-starting pressure ratio, at the 100-120 psi differential ranged from 1.7:1 to 2:1.
- Quickly closing the bypass valve at or before an assisted start did not work any better than other methods.
- Gradually increasing refrigerant pump speed (and therefore refrigerant flow) from zero did not work any better than other methods.
- Attempting to start with 100 percent flow through the expander and slowly increasing piston pressure did not work any better than other methods.
- There were 185 single assisted start events that registered a speed greater than 1 rpm.
- The team attempted 5 continuous assisted starts which consisted of repeated tapping of bump start rods.

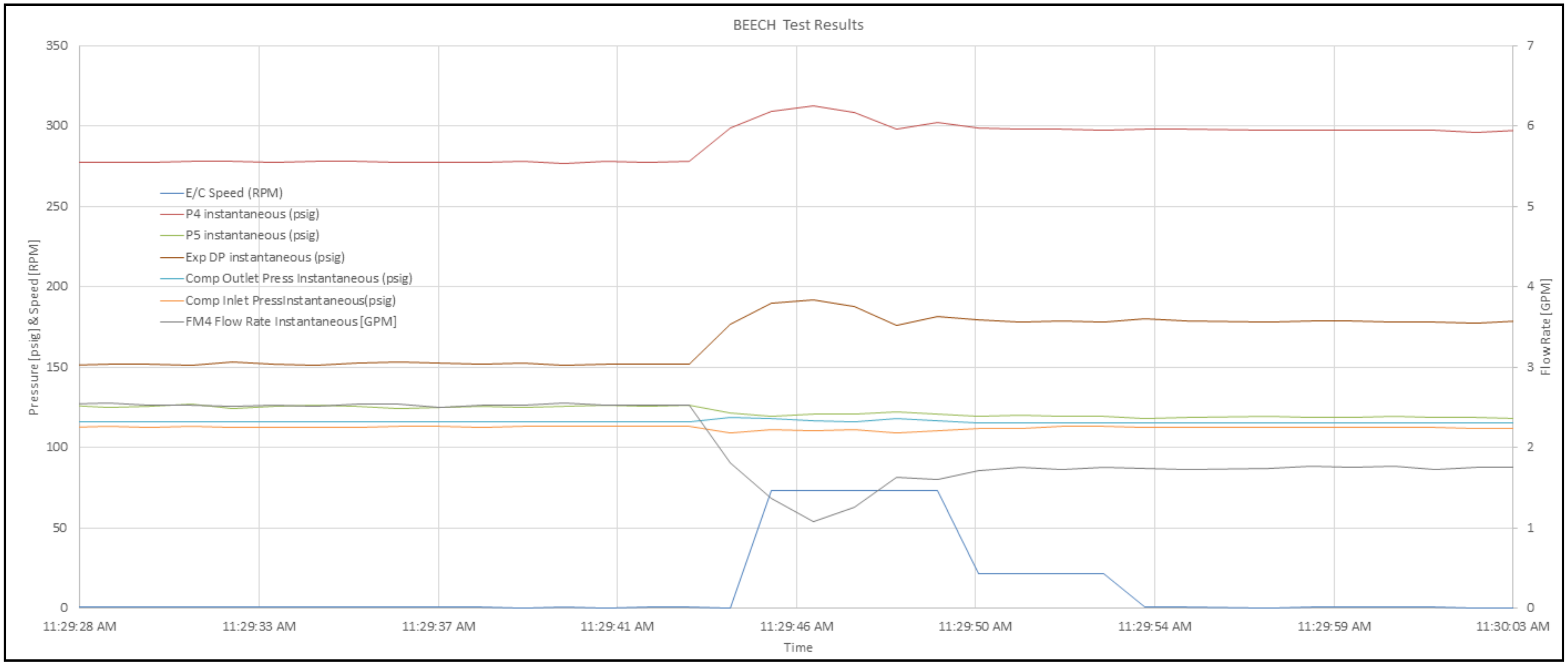
Data plots from representative start attempts are presented as Figure 63, Figure 64, and Figure 65. In these graphs, P4 is the expander inlet pressure, and P5 is the expander outlet pressure. The compressor inlet/outlet sensors, which are standard commercially-available sensors, exhibit a 2 psi offset, even when system flow is zero, and so true differential pressure can only be assumed when the measured differential pressure is more than 2 psi.

Figure 63 shows a start attempt with the bypass valve completely closed. There is a small range of positions where the expander scroll exhibits large internal leakage, allowing 100 percent of the flow to be routed through the expander. Prior to the start attempt, the expander flow was 2.5 gpm and pressure differential was 150 psi. After initiating the assisted start, the scrolls sealed and the differential pressure increased quickly, but the scroll did not move. Since it was a failed start attempt, the flow rate quickly dropped as well. When a start attempt was unsuccessful, Altex engineers quickly opened the bypass valve to avoid dead-heading the pump and having to re-prime.

Figure 64 shows an operating condition near the best-case scenario as described previously. Expander differential pressure was 125 psi, and piston pressure was 25 psi. After one assist, the unit was estimated to have completed three revolutions before stopping.

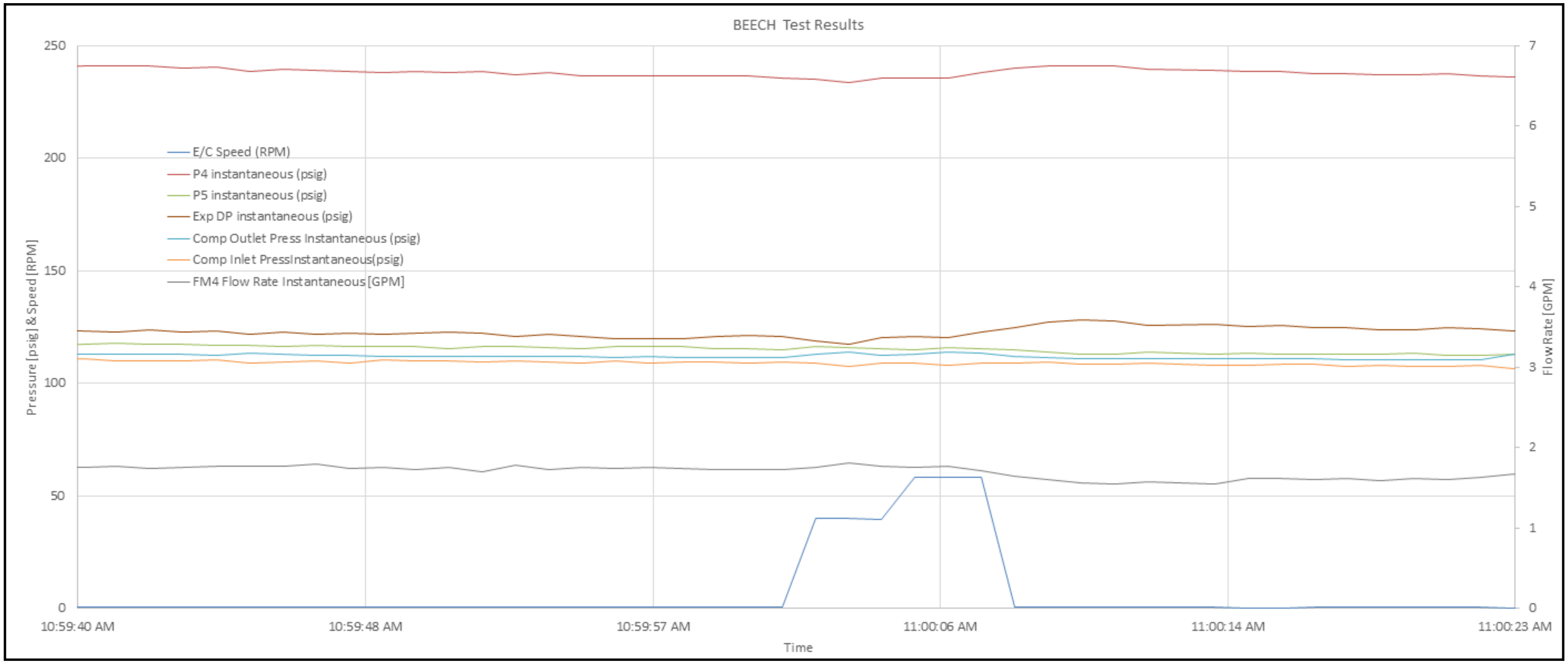
Finally, Figure 65 shows the effect of continuously assisting the start-up, by pushing on the compressor scroll multiple times, while the unit was still rotating. In the first attempt (around 1:52 PM), the bypass valve was closed slightly, while pushing on the scroll, to recover pressure. In the second attempt (after 1:53 PM), the bypass valve was maintained in a fixed position for the duration of the test. Once refrigerant started to flow through the expander, the system resistance decreased, which caused an increase in the flow rate and a decrease in pressure differential.

Figure 63: Bypass Valve Closed at Start-up, Single Mechanical Assist Event



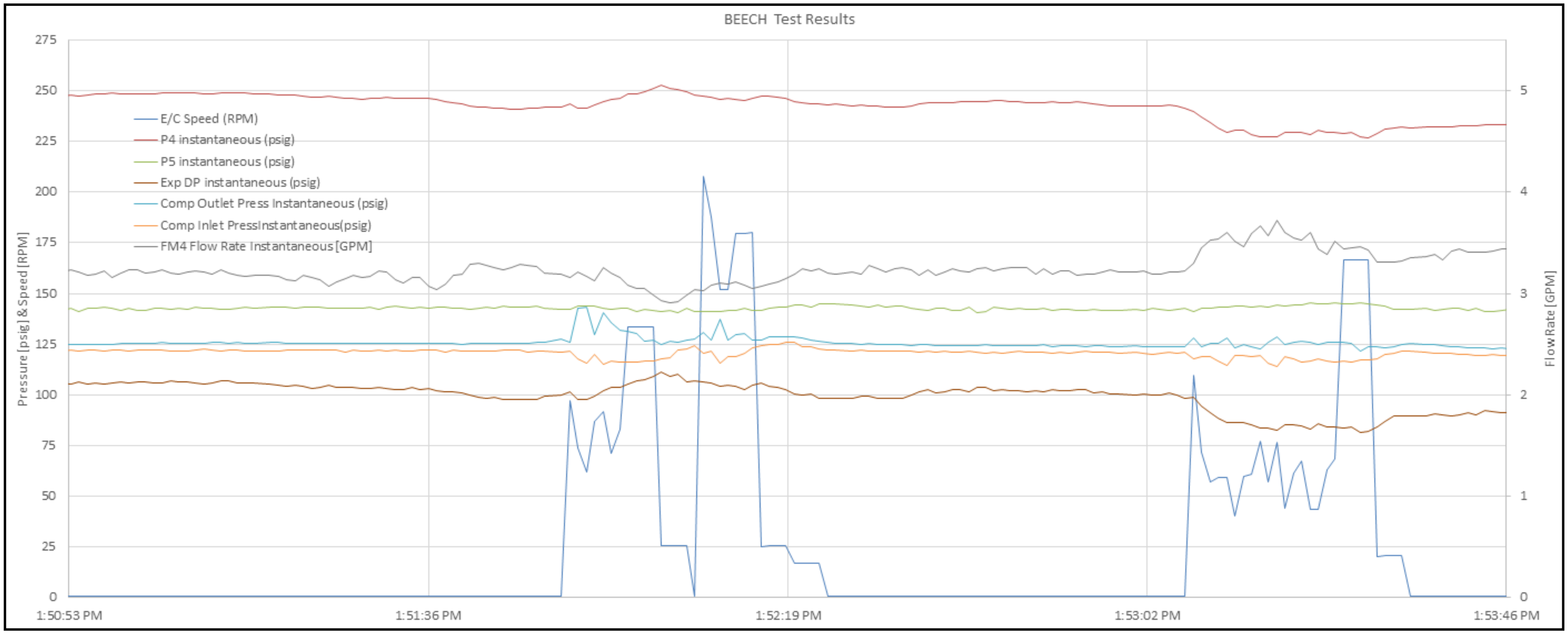
Source: Altex Technologies Corp.

Figure 64: Bypass valve partially open, 125 psi expander differential pressure, 25 psi piston pressure, and single mechanical assist



Source: Altex Technologies Corp.

Figure 65: Continuous Mechanical Assist Events



Source: Altex Technologies Corp.

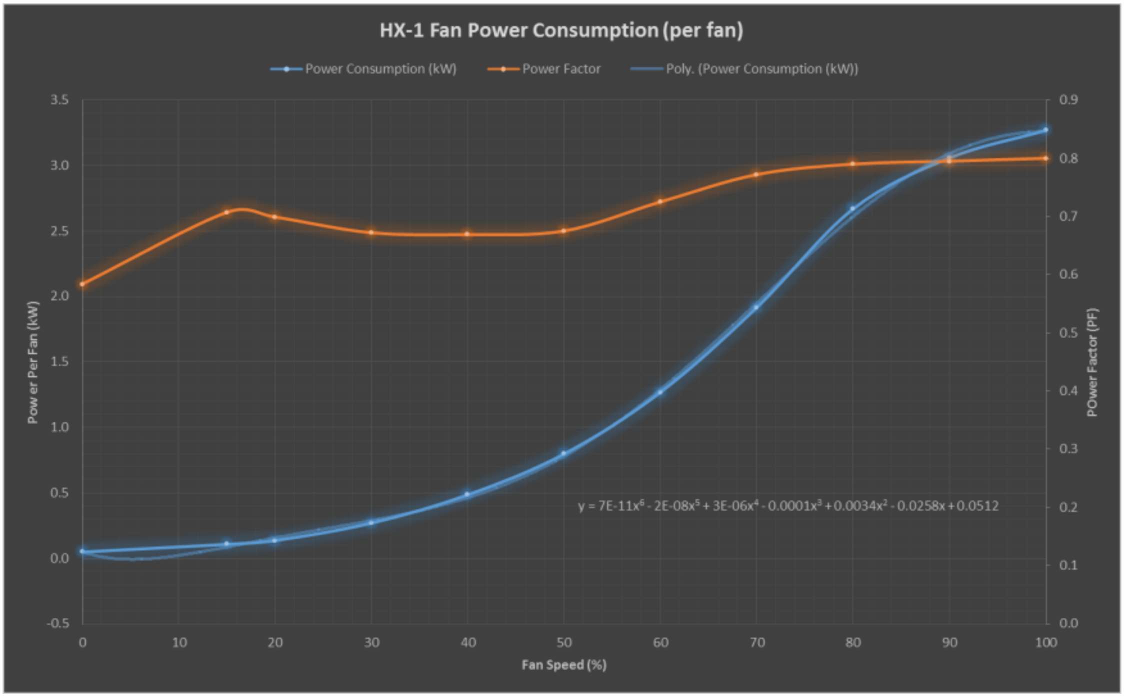
Additional Testing

While the expander/compressor testing was ongoing, testing of most other system components was completed. These tests can be divided into electrical consumption tests (to quantify power consumption of various components, which will be used in system efficiency calculations) and hot water production capability.

Electrical Power Consumption

Figure 66 shows the power consumption of the condenser fans, of which there are two in the system. These are controlled by a variable speed controller which is built into the purchased fans. Therefore, the power is shown as a function of the percentage of maximum speed, since this is the control input to the device. The test was performed using a Yokogawa WT230 meter. This device is a three phase digital power meter capable of measuring instantaneous voltage, current, power factor, and power consumption. This type of meter is required to determine the power factor over a motor speed range. This variable power factor can then be used to calculate the real power draw. Using a single, full speed power factor throughout the speed range would produce erroneous results. For reference, the power factor curve is also shown in this graph.

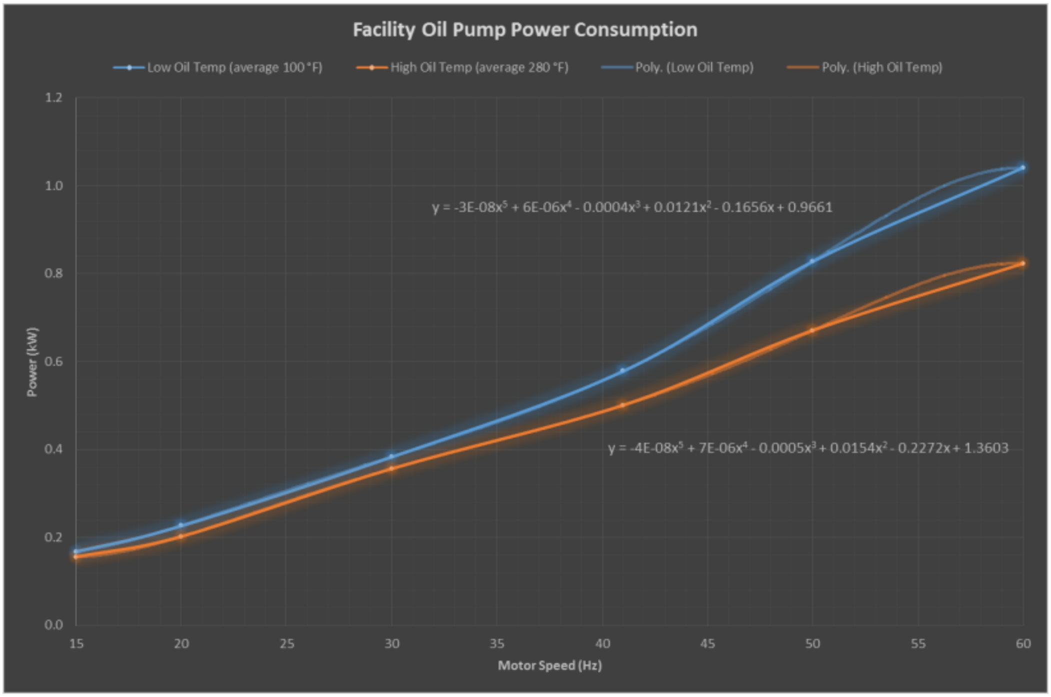
Figure 66: Measured Power Consumption, Condenser Fans



Source: Altex Technologies Corp.

Figure 67 shows the power consumption of the thermal oil pump, as installed in the system, and operating with Therminol 55, a synthetic mineral oil. The typical start-up procedure for the thermal oil system is to briefly heat the oil in the oil tank to 100˚F with an electrical resistance heater (to decrease the fluid's viscosity), and then begin pump operation. During system operation at design capacity, the oil entering the pump is expected to be 250˚F, based on 500˚F waste heat and typical generator and hot water heat exchanger operating temperatures. Therefore, the oil pump power consumption was measured at 100˚F and at 280˚F, to quantify two bounding cases. The power consumption is expressed as a function of pump speed, as measured in hertz, since the pump is controlled by a variable frequency drive (VFD), which can be adjusted by the BEECH control system to match the flow rate to the needs of the BEECH process. Power was measured using the Yokogawa meter.

Figure 67: Measured Power Consumption, Thermal Oil Pump

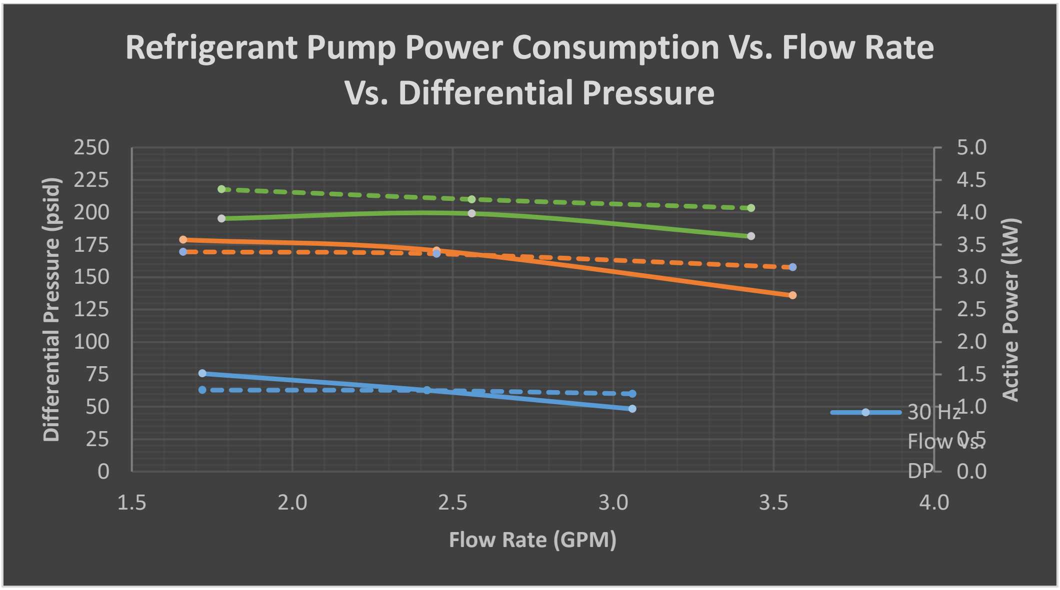


Source: Altex Technologies Corp.

Figure 68 shows the measured power consumption of the Speck-brand refrigerant pump. Like the thermal oil pump, it is controlled by a variable speed drive, and so it was characterized at three speeds, and in this case at varying differential pressures.

As noted previously, the original system design was based on a diaphragm pump, and one was used in the subcomponent testing, but this pump was very unreliable with the refrigerant of interest. One pump was found to be defective, and its replacement also performed poorly and would not prime. The manufacturer claimed that the issue was the small displacement of that model, and that the model specified for the full scale system would not have similar priming problems. To mitigate risk for the full scale system, Altex procured two pumps: one Speck-brand multi-stage, side channel pump, rated for refrigerant service, and another diaphragm pump, provided on a free-trial basis from that manufacturer as a gesture of good faith. The multi-stage Speck pump was bench tested and showed reliable operation with no priming issues.

Figure 68: Measured Power Consumption, Speck Refrigerant Pump



Source: Altex Technologies Corp.

The operating point (pressure and flow) of the BEECH system was in a very inefficient area of that pump's operating range, but the team decided to build the waste heat system with the less efficient but more reliable pump, and then later test the diaphragm pump, once expander/compressor operation was proven. As the expander/compressor starting issue persisted, diaphragm pump testing became a low priority, and the unit was returned. Table 9 shows the measured power consumption of the Speck pump, with the highlighted row being very near the intended BEECH full-scale operating point.

Table 9: Power Consumption for Speck Multi-stage Pump (Measured @ Altex)

Speed (Hz)	Flow Rate (GPM)	Pump In (psi)	Pump Out (psi)	Pump DP (psi)	V-5 (% open)	Voltage (V)	Current (A)	Power Factor	Power/phase (kW)	Power (kW)
30	3.06	120.8	169.1	48.3	46%	210.8	4.87	0.390	0.400	1.200
30	2.42	112.5	175.2	62.7	35%	210.7	4.75	0.390	0.417	1.251
30	1.72	102.1	177.7	75.6	26%	210.3	5.05	0.396	0.419	1.257
45	3.56	133.7	269.5	135.8	35%	210.4	10.37	0.480	1.050	3.150
45	2.45	115.5	286.0	170.5	25%	209.6	11.15	0.480	1.120	3.360
45	1.66	102.7	281.5	178.8	15%	209.8	11.02	0.480	1.130	3.390
50	3.43	149.1	330.5	181.4	31%	208.4	13.01	0.498	1.355	4.065
50	2.56	133.2	332.2	199.0	24%	208.5	13.35	0.505	1.400	4.200
50	1.78	116.6	311.8	195.2	15%	209.0	13.44	0.508	1.451	4.353
52	***3.42***	***149.1***	***346.6***	***197.5***	***30%***	***208.6***	***13.76***	***0.517***	***1.495***	***4.485***

Source: Altex Technologies Corp.

Table 10 shows the diaphragm pump's rated power consumption, with the highlighted row again representing the best estimation of the BEECH operating point. The potential for efficiency improvement—reducing power consumption from 4.5 kW to less than 1 kW—is substantial. The 4.5 kW would be unacceptable in a commercial product, and would result in a system with total electrical power consumption greater than that of the 5-ton chiller it was meant to improve upon.

Table 10: Power Consumption for Diaphragm-based Pump (Manufacturer Data)

Speed (rpm)	Flow (gpm)	Pressure (psi)	Motor Power (hp)	Motor Power (kW)
100	0.34	500	0.140	0.10
200	0.89	500	0.353	0.26
300	1.44	500	0.565	0.42
400	1.99	500	0.777	0.58
500	2.54	500	0.989	0.74
600	3.09	500	1.201	0.90
630	***3.26***	***500***	***1.265***	***0.94***
700	3.64	500	1.413	1.05
800	4.19	500	1.626	1.21
900	4.74	500	1.838	1.37
1000	5.29	500	2.050	1.53
1100	5.84	500	2.262	1.69
1200	6.39	500	2.474	1.85
1300	6.94	500	2.686	2.00
1450	7.77	500	3.005	2.24

Source: Altex Technologies Corp.

Hot Water Production

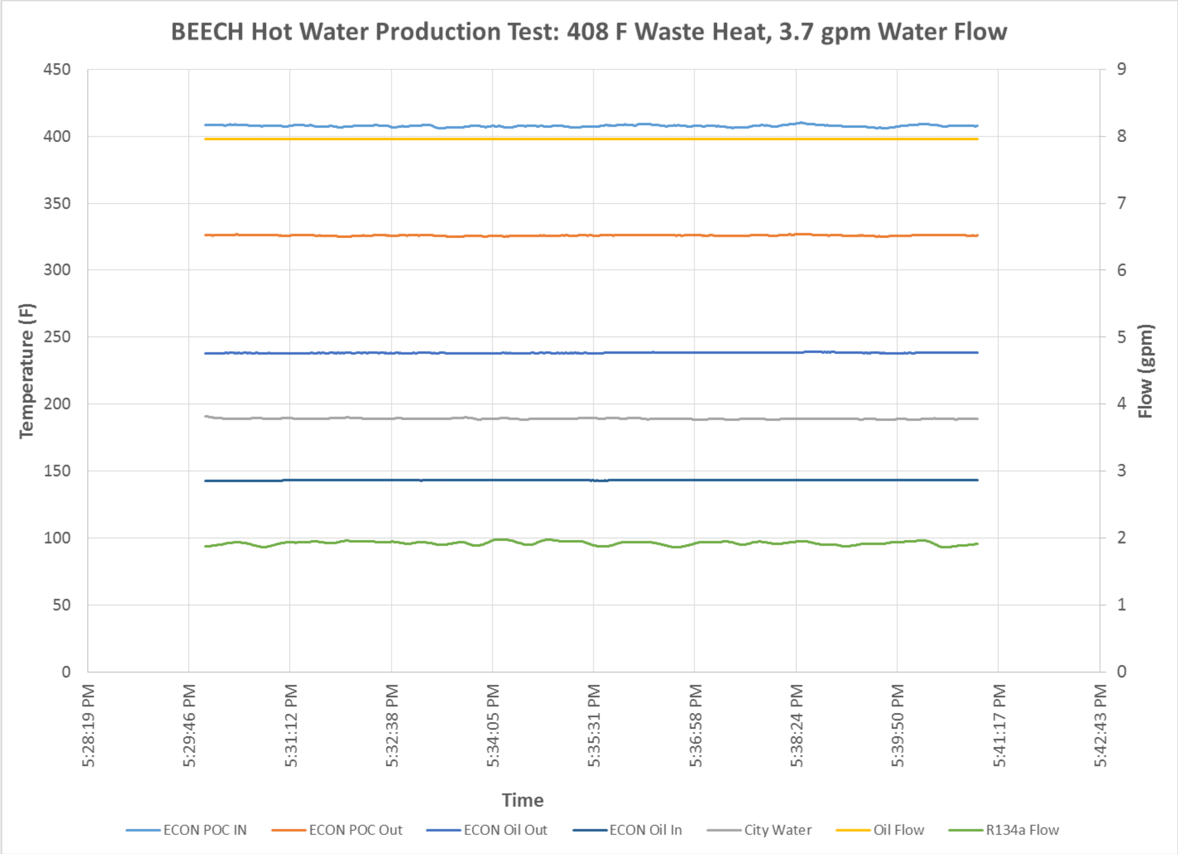
When the BEECH system is operating at full heating and cooling capacity, water is heated in two heat exchangers—one that functions as a condenser in the refrigeration circuit, and one that recovers heat from the thermal oil, after it has passed through the generator. These two heat exchangers have approximately equal heat transfer duty of 21-23 kW (maximum). Since the refrigeration circuit was not operational, data could only be taken from the thermal oil/hot water heat exchanger.

While this operating condition does not demonstrate maximum system output, it does show potential for an alternate operating mode for BEECH, or a simpler derivative system, where low temperature (200-300°F) waste heat is used to heat city water or groundwater. If expander/compressor troubleshooting is successful in later tests, the refrigeration cycle will be operated, and the increased hot water production will be used to show performance to the project goals.

For these tests, the BEECH system was operated in expander bypass mode. The refrigerant pump, generator, and air cooled condenser in the power cycle were all operated, which created representative temperatures of the oil entering the hot water heat exchanger. On the water side, water was supplied from the modified facility chiller at a regulated temperature of 64 +/- 1 ˚F, which is representative of California groundwater. To create different waste heat temperatures, the boiler thermal input was modulated. The boiler exhaust was sourced from the first pass of the boiler, where its temperature is approximately 1300˚F at these thermal inputs, and was then diluted to create typical waste heat temperatures. Oil flow was kept constant at the nominal operating condition of 8.0 gpm, and performance was tested at water flow rates of 3.7 gpm (system nominal) and 6.0 gpm.

After any change in test conditions, the system was allowed to stabilize before data was considered valid for analysis. Stability was judged by monitoring oil temperature in and out of the heat recovery heat exchanger, water temperature in and out of the hot water heat exchanger, and water flow rate. Figure 69 shows a representative plot of a steady state test point. The test results are presented in Table 11.

Figure 69: Sample Steady-State Hot Water Test Data Point



Source: Altex Technologies Corp.

Table 11: Hot Water Generation Test Results

Boiler Thermal Input (MMBtu/hr)	Waste Heat Temperature (°F)	Therminol 55 Flow (gpm)	Therminol T @ HX Inlet (°F)	Hot Water Flow (gpm)	Hot Water Inlet T (°F)	Hot Water Outlet T (°F)
1.95	408	8.0	167	3.7	63.7	83.4
1.95	409	8.0	168	6	64.2	77.3
2.35	499	8.0	194	3.7	64.0	89.7
2.35	501	8.0	188	6	64.3	80.0
3.07	628	8.0	174	3.7	63.4	84.6
3.07	629	8.0	175	6	63.5	77.4

Source: Altex Technologies Corp.

Testing Conclusions

As the scope and challenges of the BEECH system have evolved throughout the project, startup operation of the novel expander/compressor was the most obvious challenge. Under Task 5 activities, Altex, with cooperation of minor subcontractor Legacy Chiller Systems, performed extensive mechanical modifications and test sequences to attempt diagnosis and resolution of the starting issues. More than 200 tests were performed, using both R-134a and pressurized nitrogen as the working fluid.

Meanwhile, the team did complete electrical power consumption measurements, and performed a limited test of hot water generation capabilities. These data sets are used the next chapter to determine performance towards the project goals of natural gas avoidance, reduction in pollutant and greenhouse gas precursor emissions, and projected system operating cost and payback times. The projected full-system performance to those goals is also being calculated, based on the system process model.

CHAPTER 11: Evaluation of Results

The final BEECH process is based on scroll technology and has five tons cooling and 3.7 gpm hot water nominal outputs. This updated design was presented as part of the successful Critical Project Reviews. Table 12 presents the project goals, as revised to match the new system design.

Table 12: Project Goals

Goal	Initial Project Goal	Revised Design	Comment
Cooling Output (Btu/hr)	180,000	60,000	Based on CPR #2
Avoidance of Electrically-driven Cooling Costs (kW-h/day)	400	133	Scaled from new cooling output goal
Heating Output (Btu/hr)	360,000	190,000	Based on CPR #2
Thermal Efficiency Improvement (%)	10%	10%	Basis of comparison not noted
Greenhouse Gas Reduction (lbs/day)	1010	533	Scaled from new thermal output goal
Natural Gas Reduction (%)	7.5%	7.5%	Basis of comparison not noted
Payback Time, Waste Heat (yrs)	2	2	No change
Payback Time, Solar Thermal (yrs)	5	5	No change

The comparison basis for the percent improvement in thermal efficiency and natural gas reduction was not defined in the Statement of Work. The amount of waste heat available to be converted into useful outputs by BEECH will depend on the thermal input and thermal efficiency of the equipment to which it is mated, and also how that equipment is operated. Therefore, as part of this report, the equipment size and its associated waste heat will be defined, on the basis of the 7.5 percent natural gas reduction goal and the amount of hot water produced. The 10 percent "thermal efficiency improvement" is very closely related to the reduction in natural gas consumption, since 190,000 Btu/hr of the total 250,000 Btu/hr system output is thermal output.

Since operation of the expander/compressor was not demonstrated, the system evaluation uses a combination of experimental data and projected results based on the CHEMCAD process model.

Cooling Output-based Goals

Since cooling output was not produced, cooling-based analyses are based on the CHEMCAD model prediction of 5 tons (60,000 Btu/hr) cooling output. The output of the BEECH system would partially displace the output of a central Mechanical Vapor Compression (MVC) chiller. For comparison, the Trane CGAM line of chillers[7] was chosen, which has a nominal Coefficient of Performance of 3.5, meaning that the cooling output is 3.5x that of the electrical power input. The power consumption of the Trane unit to produce 60,000 Btu/hr is:

$$Power\ consumed\ per\ day = \frac{60{,}000\,\frac{btu}{hr}}{3.5} * \frac{1\ kw}{3416\,\frac{btu}{hr}} * \frac{24\ hrs}{day} = 120\ kWh/day$$

The associated accounting of greenhouse gas reduction, due to avoidance of electrically-driven cooling, is then calculated as[8]:

$$CO_{2\ avoided} = 0.83\,\frac{lbs}{kWh\ saved} *\ 120\,\frac{kWh}{day} = 99.6\,\frac{lbs}{day}$$

The assumption of 24 hour/day operation is based on the system sizing performed in Site Specification activities, which indicated that most buildings, with a sufficient waste heat supply to justify the installation of BEECH, would also have a year-round base load of greater than 5.0 tons cooling. However, when considering long-term impacts of BEECH, it is reasonable to assume some downtime for non-typical conditions and general maintenance.

Heating Output-based Goals

Water is heated in both the refrigeration cycle condenser and in HX-3, the heat exchanger downstream of the generator. As discussed in the Site Specification Report, the bacteria that causes Legionnaire's Disease can multiply in stagnant water under 115 ˚F (46 ˚C), so the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) recommends a minimum hot water storage temperature of 140 ˚F (60 ˚C). The maximum water usage temperature identified by ASHRAE is 194˚F (90˚C), for dish rinsing applications. The median point of those two temperatures, 167˚F, was chosen as the BEECH design point. The theoretical maximum hot water production, based on a 62˚F groundwater temperature is therefore:

$$Hot\ Water\ Energy = 3.7\,\frac{gallons}{minute} * (167F - 62F) * 8.3\,\frac{\mathrm{lb}}{\mathrm{gal}} * 1\,\frac{btu}{lb - F} * 60\ \frac{min}{hr} =\ 193{,}473\ btu/hr$$

Since continuous usage of the BEECH system will require storage capacity to even out transients in hot water demand, an alternative system configuration can be imagined where part of the

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7 "Air-Cooled Scroll Chillers Model CGAM - Made in USA 20-130 Nominal Tons (50 Hz and 60 Hz)." Publication # CG-PRC017-EN, Trane Corp. January 2012.

8 CO_{2e} factor sourced from PON-12-503, Appendix 17.

system thermal output would be used to heat water to the lower (140˚F) storage temperature, and then the remaining available heat would be used to maintain the storage tanks at 140˚F, thus negating thermal losses from the tanks. Since this imagined system's capacity and construction would vary based on the specific facility, analysis of that system configuration is outside the scope of this report, but is an example of the flexible ways in which BEECH could be integrated in a facility to use at least the 193,473 Btu/hr of heat calculated above.

The production of hot water replaces the output of a natural gas-fired boiler, and does so without additional natural gas consumption. The natural gas displacement is then equal to the amount of natural gas the boiler would have consumed to produce that same hot water, which must also consider the boiler's efficiency. In a facility that uses a steam boiler to produce steam, and then a secondary heat exchanger to produce hot water, additional efficiencies and losses could also be considered. A new firetube boiler with a low excess air burner can be up to 82 percent efficient, but an older boiler with a high excess air burner and secondary heat exchangers could be less than 78 percent efficient. For simplicity in calculation, a single 80 percent boiler efficiency is assumed.

$$Natural\ Gas\ Displaced = 193{,}473\frac{btu}{hr} * \frac{1}{80\%\ boiler\ eff} * \frac{therm}{100{,}000\ btu} * 24\frac{hrs}{day} = 58.0\frac{therms}{day}$$

The greenhouse gas reduction from this avoidance is calculated as[9]:

$$CO_{2\ avoided} = 11.7\frac{lbs}{therm\ saved} * \ 58\frac{therms}{day} = 679.1\frac{lbs}{day}$$

The NOx and CO reduction due to natural gas avoidance was calculated with standard United States Environmental Protection Agency (US EPA) procedures.[10] Burner emissions were assumed to be 9 ppm NOx and 50 ppm CO, corrected to 3 percent dry O_2. These limits are reflective of South Coast Air Quality Management District limits for 2-10 MMBtu/hr boilers.[11] The method was then used to calculate a conversion factor from the therms per day of natural gas displaced to pounds of NOx and CO avoided per day:

$$CO_{\ avoided} = 58\frac{therms}{day} * 0.0037\frac{lbs}{therm} = 0.214\ \frac{lbs}{day}$$

$$NOx_{avoided} = 58\frac{therms}{day} * 0.0011\frac{lbs}{therm} = 0.0634\ \frac{lbs}{day}$$

More details on these calculations, including the standard factors used, are included on the worksheets that are presented in Appendix B.

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9 CO_{2e} factor sourced from PON-12-503, Appendix 17.

10 "Output-Based Regulations: A Handbook for Air Regulators" US EPA. August 2004, page 26.

11 Ref. SCAQMB Rule 1146: http://www.aqmd.gov/docs/default-source/rule-book/reg-xi/rule-1146-1.pdf?sfvrsn=4.

The results of actual hot water tests using only the hot oil-water heat exchanger were previously presented. The 500˚F/ 3.7 gpm data point is considered the most-typical point for both the waste heat temperature and the water flow rate. In the ~628˚F testing, the refrigerant pressure and flow rates were not representative of typical operating conditions, and so the heat transfer results were lower than expected.

The original project goals included system performance testing with waste heat at temperatures of 200-300˚F (93-149˚C). The final design of the BEECH cycle integrates two thermodynamic cycles, both operating on R-134a. The refrigeration cycle's lower temperature limit is bounded by condenser water temperature. With typical city or ground water temperatures of 62˚F (17˚C), the condenser temperature will be at least 70˚F (21˚C), or higher. This is not an adequate temperature differential from a 200˚F waste heat temperature to effectively accomplish the remainder of the BEECH processes, to generate both heating and cooling outputs at financially-viable scales. However, a 200-300˚F (93-149˚C) waste heat source can be used to generate the required 140 or 167˚F (60 or 75˚C) hot water. This condition is reflected in Table 11, where an oil/water oil heat exchanger with 167-194˚F (75-90˚C) oil inlet temperature is heating water. Practically speaking, a facility would be better served by implementing a more-simple system, consisting of the heat recovery heat exchanger and directly heating the water in that heat exchanger, without the added complexity of the intermediate oil loop.

Considering now the data of Table 11 in relation to system performance testing at 400-600˚F (204-316˚C), the natural gas displaced (based on this limited test) can be determined, using the same calculation methods as used above. The results are shown in Table 13:

Table 13: Hot Water Generation Test Results

Test ID	Boiler Thermal Input (MMBtu/hr)	Hot Water Flow (gpm)	Hot Water Inlet T (°F)	Hot Water Outlet T (°F)	NG Avoidance (therms/day)	NG Reduction (% of input)
1	1.95	3.7	63.7	83.4	11.2	2.4%
2	1.95	6	64.2	77.3	11.8	2.5%
3	2.35	3.7	64.0	89.7	14.3	2.5%
4	2.35	6	64.3	80.0	14.2	2.5%
5	3.07	3.7	63.4	84.6	11.9	1.6%
6	3.07	6	63.5	77.4	12.6	1.7%

Source: Altex Technologies Corp.

The data shown in Table 11 and Table 13 was produced using a single oil/water heat exchanger, and with the 10 MMBtu/hr boiler operating at only 20-30 percent of rated input. As expected, under these conditions where the full system was not operated, the predicted 58 therms/day natural gas avoidance of the full system was not achieved.

However, a natural gas reduction is still calculated, based on the actual thermal input to the boiler. As shown in Appendix B, Altex engineers calculated mass flow through the burner by measuring natural gas flow (via the facility's utility meter) and stack oxygen percentage (using a calibrated emissions analyzer). This information was used to back-calculate air and gas mass flow, using tabular values for densities. To accurately determine the total mass flow through the heat recovery heat exchanger (which includes the exhaust plus the dilution air), the specific heat (cP) of the exhaust must be calculated, since it contains sufficient water vapor to invalidate a simple dry-air approximation. Specific heat was calculated by gravimetrically weighting the specific heat of individual compounds in the products of combustion (POC),[12] based on the known oxygen concentration of the undiluted exhaust. With this known, as well as all other temperatures and the burner flow rate, the dilution air flow could be calculated, as well as the cP of the complete mixture.

The amount of heat recovered from the exhaust was calculated based on the total mass flow, specific heat of the POC/dilution air mixture, and temperature in and out of the economizer. As shown in Appendix B, heat transferred from the oil to the water was calculated using the specific heat, density, temperatures in and out, and flow rates for each fluid. The percent difference was calculated for each case, and is an indication of the thermal losses from piping and connections between the measurement points.

For data point #3, which is closest to the nominal design condition of BEECH, the total heat recovery from the exhaust was 255,502 Btu/hr, or approximately 10 percent of the total thermal input to the boiler. Since this is the maximum available heat to the BEECH system, the resulting output from BEECH, when operating with both cooling and heating functions, cannot exceed this input, due to inefficiencies and minor losses. Therefore, this particular test cannot demonstrate the 10 percent thermal efficiency improvement target. When the system is fully operational, more heat can be extracted from the thermal oil, and system useful output will increase.

The minimum size of thermal equipment for which the BEECH system will produce a 7.5 percent reduction in natural gas usage can be predicted from the modelled hot water output. The thermal equipment must meet two criteria: it must have enough waste heat available to power the BEECH system, but not be so large that the 193,000 Btu/hr (1.93 therms/hr) replaces less than 7.5 percent of its input. For example, an 80 percent efficient boiler with 30 therms/hr input has 6 therms/hr of waste heat, as shown in Table 14, of which enough can be recovered to power BEECH. The produced 1.93 therms (193,000 Btu/hr) of hot water replaces 2.41 therms of input into the boiler, accounting for the boiler's efficiency, which is an 8 percent payback replacement. If this boiler was more efficient, not enough waste heat would be available (to power this capacity of BEECH), and if it was larger, the produced hot water would not replace enough of the thermal input to meet the 7.5 percent target.

The criteria for "adequate available waste heat" in Table 14 was assumed to be five therms/hr, or 500,000 Btu/hr. The capacity of BEECH used throughout these analyses requires approximately 290,000 Btu/hr heat to function at full capacity, according to the system model.

[12] The POC composition was calculated using formulas in the North American Combustion Handbook Vol 1, Third edition (equations 3.6 thru 3.9b).

This would then represent 58 percent or less waste heat recovery, which is a reasonable goal. Greater recovery is possible, depending on the desired hot water output of the system, as well as the type of heat recovery heat exchanger and the water vapor content of the waste heat. For example, a dry heat source (which presents no water condensation issues) of 500˚F (260˚C) can be cooled to 150˚F (66˚C), if the desired hot water temperature is 140˚F (66˚C), resulting in an 81 percent heat recovery.

Table 14: Potential Therms of Available Waste for Equipment with Various Inputs and Efficiencies

Thermal Efficiency	Total Thermal Input, therms/hr 10	15	20	25	30	35	40
50%	5.0	7.5	10.0	12.5	15.0	17.5	20.0
55%	4.5	6.8	9.0	11.3	13.5	15.8	18.0
60%	4.0	6.0	8.0	10.0	12.0	14.0	16.0
65%	3.5	5.3	7.0	8.8	10.5	12.3	14.0
70%	3.0	4.5	6.0	7.5	9.0	10.5	12.0
75%	2.5	3.8	5.0	6.3	7.5	8.8	10.0
80%	2.0	3.0	4.0	5.0	6.0	7.0	8.0
85%	1.5	2.3	3.0	3.8	4.5	5.3	6.0
90%	1.0	1.5	2.0	2.5	3.0	3.5	4.0

(green shaded cells indicate adequate waste heat available for 5 tons/190KBtu/hr BEECH)

Source: Altex Technologies Corp.

Table 15 shows the same range of equipment analyzed in Table 14, and calculates the input required to generate 1.93 therms, for each efficiency, and then shows what natural gas displacement percentage would be caused by that generation. The resulting green-shaded cells therefore represent the range of facilities where a BEECH system with 193,000 Btu/hr of cooling output could replace at least 7.5 percent of its the natural gas input. This range of equipment capacities and efficiencies is consistent with the equipment found in many mid-sized commercial buildings.

Evaluation of Electrical Consumption

The section on cooling output-based goals described the avoidance of electrically-driven cooling costs, and the associated reductions in greenhouse gas precursors and pollutants. However, some components of the BEECH system need electricity to operate, which partially offsets the cooling cost savings. Under Task 5 activities, the power consumptions of the various system components were measured during testing, and are summarized in Table 16.

The CHEMCAD model assumed efficiencies for the various components, based on industry data. The greatest discrepancy is seen for the HX-1 condenser fans. Altex staff consulted with subcontractor Legacy Chillers to specify a high efficiency, state-of the-art commercial condenser/fan unit with variable speed fans, so the higher consumption value was a surprise to researchers. The CHEMCAD model assumed a heat exchanger with 0.5 in WC pressure drop, and fans producing 80,000 cfm air flow. The purchased unit operates with only 23,000 cfm, but the manufacturer does not publish heat transfer area or pressure drop ratings, and so it is difficult to compare its performance to the model's assumptions. Future commercialization activities should therefore include market research on other models/brands of commercial condenser/fan packages with lower fan power consumption, to meet or improve upon the model predicted performance. Similarly, the refrigerant pump selection should be examined carefully, to ensure the specified unit meets performance and reliability requirements.

Table 15: Potential Natural Gas Replacement for Equipment with Various Inputs and Efficiencies

Thermal Efficiency	Input Required to Produce 1.93 therms/hr (therms/hr)	Total Thermal Input, therms/hr						
		10	15	20	25	30	35	40
50%	3.86	38.6%	25.7%	19.3%	15.4%	12.9%	11.0%	9.7%
55%	3.51		23.4%	17.5%	14.0%	11.7%	10.0%	8.8%
60%	3.22		21.4%	16.1%	12.9%	10.7%	9.2%	8.0%
65%	2.97		19.8%	14.8%	11.9%	9.9%	8.5%	7.4%
70%	2.76			13.8%	11.0%	9.2%	7.9%	6.9%
75%	2.57			12.9%	10.3%	8.6%	7.4%	6.4%
80%	2.41				9.7%	8.0%	6.9%	6.0%
85%	2.27						6.5%	5.7%
90%	2.14							

All cells with data indicate adequate waste heat available for 5 tons/190KBtu/hr BEECH; green shaded cells indicate potential for >7.5% NG replacement.

Source: Altex Technologies Corp.

Table 16. Power Consumption, Modelled and Measured

Component	CHEMCAD Process	Measured
HX-1 Fans	1.61 kW	3.84 kW[13]
Facility Oil Pump	0.6 kW	0.5 kW
Refrigerant Pump	0.63 kW	0.9 kW[14]
Miscellaneous	---	0.1 kW[15]
Total Electrical	2.84 kW	5.34 kW
Total Elec.	68.2 kWh/day	128.2 kWh/day

Source: Altex Technologies Corp.

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13 HX-1 fan power consumption was based on 70% fan speed data, as presented in the Task 5 report.

14 Based on manufacturer data for a diaphragm pump. Actual power consumption measured with the high output, multi-stage pump used in the test system was 4.1 kW. Further details are available in the Task 5 report.

15 Assumed miscellaneous power consumption for controls, solenoid valve drivers, and touchscreen display.

The net potential decrease in electrical power consumption with BEECH is therefore 51.8 kWh/day (120 kWh/day – 68.2 kWh/day), and the net carbon dioxide (CO_2) reduction would be:

$$CO_{2\,avoided} = 0.83\frac{lbs}{kWh\ saved} * \ 51.8\frac{kWh}{day} = 43\frac{lbs}{day}$$

Payback Time Assessment

To evaluate the system payback time, for solar or waste heat BEECH, an equivalent alternative had to be established as a baseline for comparison. Since the BEECH output is designed to be less than the base load of a facility, the alternative system was assumed to be already installed and depreciated (or for new installations, installed as a parallel redundant system), and consist of a chiller of 5 tons' capacity or greater, and a natural-gas fired boiler of greater than 241,000 Btu/hr thermal input (yielding a >193,000 Btu/hr output at 80 percent efficiency). For the commercial buildings identified in Site Specification, these capacities are reasonable. The boiler is actually ensured to have a much greater thermal input, to generate enough waste heat to supply BEECH, as calculated in Table 14.

Operating costs were calculated using both the standardized utility price parameters, as well as updated 2015 utility costs. Details of the 2015 utility cost calculation are included in Appendix C. BEECH uptime at full capacity was assumed to be 3000 hours/year for the solar thermal version, and 8000 hours/year for the waste heat version.

BEECH system prices are based on the Bill of Materials (BOM) and fabrication/assembly costs calculated in the next chapter. The total prices for the heat recovery heat exchanger and solar thermal subsystems have been updated based on actual costs (for example, a 2015 quote from Kingspan Solar, updating the previous 2013 quote), and 20 percent allowances have been included for installation costs of the solar thermal collectors, heat recovery heat exchanger, and base BEECH subsystems, to reflect the full capital expenditure. Actual installation costs will, of course, vary greatly by facility.

Finally, an alternative basis for comparison to the waste heat system is presented, consisting of only the heat recovery heat exchanger and the associated controls and circulation system, with 20 percent additional for installation. Since the majority of BEECH's output value is in the hot water output, this simpler system is a lower capital cost option for facilities not interested in reducing their electric load, or for whom the payback times below are not acceptable. Similar simplifications to the solar thermal system are well known and promoted by manufacturers and installers of solar thermal collectors.

Table 17 summarizes the analysis's assumptions (with additional detail available in Appendix C); Table 18 and Table 19 present the results.

Table 17: Summary of Payback Analysis Assumptions

Annual Operating Hours	Solar	3000 hrs
	Waste Heat	8000 hrs
5 Ton Chiller	COP	3.5
Hot Water Calculation	Boiler Efficiency	80%
	Water Flow Rate	3.7 GPM
	Differential Temp.	110 °F
	Specific Heat	1 Btu/lbm·°F
Electricity Price	Solicitation	$0.13/kWh
	2015	$0.17/kWh
Natural Gas Price	Solicitation	$0.68/therm
	2015	$0.71/therm
Solar Incentives	Natural Gas	$20.19/therm (installed capacity)
	Electricity	N.A.

Source: Altex Technologies Corp.

Table 18: Payback Summary, 2015 Utility Pricing

	Initial Capital Cost (before Incentives)	Actual Capital Cost (with Incentives)	Operating Cost/year	Net Operating Cost Savings/year	Payback Time (years)
BEECH Solar	$150,317	$83,629	$1,448	$6,507	12.9
BEECH Waste Heat	$69,904	$69,904	$3,862	$17,353	4.0
HRHX + Chiller	$21,122	$21,122	$6,825	$14,391	1.5
Solar Equivalent	$0	$0	$7,956	N.A.	N.A.
Waste Heat Equivalent	$0	$0	$21,216	N.A.	N.A.

Source: Altex Technologies Corp.

Table 19: Payback Summary, 2012 Solicitation Utility Pricing

	Initial Capital Cost (before Incentives)	Actual Capital Cost (with Incentives)	Operating Cost/year	Net Operating Cost Savings/year	Payback Time (years)
BEECH Solar	$150,317	$83,629	$1,108	$6,018	13.9
BEECH Waste Heat	$69,904	$69,904	$2,954	$16,048	4.4
HRHX Only	$21,122	$21,122	$5,219	$13,783	1.5
Solar Equivalent	$0	$0	$7,126	N.A.	N.A.
Waste Heat Equivalent	$0	$0	$19,002	N.A.	N.A.

Source: Altex Technologies Corp.

Remaining Assessments

The remaining assessments identified in the introduction are addressed below. In some cases, the lack of full system test data necessitates a subjective evaluation.

Solar Thermal System Performance Under Varying Insolation Levels

To evaluate the system performance at varying solar insolation levels, the assumed 30-panel Kingspan array was examined using resources from the National Renewable Energy Laboratory (NREL). As shown in the calculations below, December output is 44 percent lower than in June. Since turndown analyses could not be completed with an operating system, the effect of lower thermal input was not tested. However, it is likely that this large decrease in input energy would result in such a low cooling output that it would be more worthwhile to operate BEECH to produce only hot water, especially since facility cooling needs will be lower in the winter months.

DF 100 (30 tubes) Model Data[16]

- Aperture Area: 3.23 square meter
- 30-collector array: 96.9 square meters or 1043 square feet
- Efficiency[17] at 143˚F fluid outlet/inlet temperature differential: 0.66
  $Efficiency = -0.00000417 T_{collector-ambient}{}^{2} - 0.00032778 T_{collector-ambient} + 0.79375556$

Calculation for June

- Direct normal solar resource available for California in June: 8 kWh/m2 or 2536 Btu/ft2 per day.[18]

$$Total\ radiation\ on\ 30\ collector\ array = 2536\ \frac{Btu/_{ft^2}}{day} \times 1043\ ft^2 = 2{,}645{,}048\ \frac{Btu}{day}$$

- BEECH Solar Output: 1.746 MMBtu/day or 193,970 Btu/hr:

$$BEECH\ Solar\ Output = 2{,}645{,}048 \frac{Btu}{day} \times 0.66 \times \frac{day}{9\ hr} = 193{,}970\ {}^{Btu}/_{hour}$$

Note: Task 3 designed a 30 panel array, based on standard panel efficiencies and their rated 10,000 Btu/hr peak output. This updated June calculation, using panel efficiencies for high differential temperatures, indicates that the 30 panel array may be slightly undersized, though this calculation and the solar test calculation are based on average and peak outputs, respectively, so some discrepancy can be accepted.

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16 Kingspan Solar – DF100 : http://www.kingspansolar.com.ua/sites/default/files/documents/DF100.pdf.

17 Kingspan Efficiency Spreadsheet for DF100 at Elevated Temperatures, direct e-mail from Kingspan Solar Engineering to Altex Technologies.

18 NREL.org – Concentrating Solar Power Radiation Map (June): http://www.nrel.gov/gis/images/map_csp_us_10km_june_feb2009.jpg.

Calculation for December

- Direct normal solar resource available for California in June: 4.5 kWh/m2 or 1426 Btu/ft2 per day:[19]

$$Total\ radiation\ on\ 30\ collector\ array = 1426\ \frac{Btu/_{ft^2}}{day} \times 1043\ ft^2 = 1{,}487{,}318\ \frac{Btu}{day}$$

- BEECH Solar Output: 0.982 MMBtu/day or 109,070 Btu/hr:

$$BEECH\ Solar\ Output = 1{,}487{,}318 \frac{Btu}{day} \times 0.66 \times \frac{day}{9\ hr} = 109{,}070\ ^{Btu}/_{hour}$$

System performance under conditions of transient cooling demand, and domestic hot water and process make-up water demands. Collect data on actual uptime and seasonal utilization of BEECH heating and cooling activities.

Site Specification activities showed that the 3.7 gpm hot water and 5.0 tons cooling will be less than the base load of the facilities of interest. Given adequate hot water storage capacity and no turndown issues for the legacy equipment, transient cooling and heating demands should not affect BEECH operation. The solar thermal input limitation noted above would limit cooling output, if a facility happened to have a high cooling demand during winter months.

Cooling and heating performance data under maximum capacity and maximum system turndown conditions

Ongoing expander/compressor issues prevented testing at maximum thermal output, or at any cooling capacity. Maximum turndown demonstrated in limited testing is indicated in Test Point #1 in Table 13, which produced 37,202 Btu/hr of hot water, which represents a 5.1:1 turndown from the maximum rated system output of 193,000 Btu/hr.

Determine mean time between failures of components and projected field maintenance costs

Ongoing expander/compressor issues prevented extended testing, and so MBTF or maintenance costs of that novel component are unknown. Teardown inspections of the unit after more than 200 attempted starts, including some with known low-oil conditions, have shown no sign of wear or degradation, which bodes well for durability. All other BEECH system components are commercial off-the-shelf parts rated for their respective working fluids, and so their durability and service costs are expected to be in-line with refrigeration and boiler-room equipment norms.

Expected air emissions reduction of oxides of nitrogen (NOx), carbon monoxide (CO), carbon dioxide (CO_2), and nonmethane hydrocarbons

The expected NOx, CO, and CO_2 reductions were determined in Section 11.2. While unburned hydrocarbons are a regulated pollutant in most air quality districts, their emission is unlikely to

[19] NREL.org – Concentrating Solar Power Radiation Map (December): http://www.nrel.gov/gis/images/map_csp_us_10km_december_feb2009.jpg.

be affected, positively or negatively, by a BEECH installation. While a reduction in natural gas usage by a device that emits hydrocarbons would theoretically reduce those emissions, the reduction would be better implemented through proper tuning and maintenance of the existing thermal equipment.

Determine peak shaving capability, due to BEECH's inherent ability, when operating with solar collectors, to generate maximum cooling capacity at peak insolation times

Modelling and data and solar thermal component analysis support BEECH's ability to peak shave during peak cooling demand times. However, the net 2.5 kW savings of the system over an equivalent 5-ton chiller represents a minimal effect on the overall grid, and is a lesser benefit of BEECH.

Summary of Analysis Results

The previous chapters have detailed the analysis of data required to show performance of the system relative to the project goals. The key goals identified in Table 1 are repeated below in Table 20 with their associated analysis results.

Table 20: Summary of Key Analysis Results

Goal	Initial Project Goal	Revised Design	Analysis Results	Comment
Cooling Output (Btu/hr)	180,000	60,000	60,000	Results based on model; Cooling output not achieved in testing
Avoidance of Electrically-driven Cooling Costs (kW-h/day)	400	133	58.1	Based on modelling results and data—net savings
Heating Output (Btu/hr)	360,000	190,000	193,473	Based on modeling results
Thermal Efficiency Improvement (%)	10%	10%	>10%	Based on modelling results, for thermal equipment of <75% efficiency
Greenhouse Gas Reduction (lbs/day)	1010	533	722	Total CO_2 avoided from electric and NG avoided, at maximum designed output
Natural Gas Reduction (%)	7.5%	7.5%	>7.5%	Achievable at full system output for equipment with 2.0-4.0 MMBtu/hr thermal input.
Payback Time, Waste Heat (yrs)	2	2	4.0	Based on 2015 utility pricing
Payback Time, Solar Thermal (yrs)	5	5	12.9	Based on 2015 utility pricing

Source: Altex Technologies Corp.

CHAPTER 12: Technology Readiness and Commercialization

Altex is a small research, development and deployment company, with more than $7 million/year in sales. Altex has supported burner manufacturers, as well as other private and government clients, in fuels combustion, emissions control, and power system developments. Clients include the United States Army, Defense Advanced Research Projects Agency, United States Navy, United States Air Force, United States Department of Energy, United States Environmental Protection Agency, California Energy Commission, California Air Resources Board, Southern California Gas, Electric Power Research Institute, ST Johnson, Eclipse Combustion, Gordon Piatt Energy Group, Cleaver Brooks, NIECO, Alzeta, and Riley Stoker. To commercialize technologies, Altex works cooperatively with manufacturers. For example, Altex has developed and tested a low-NO_x burner for boilers, which has been commercialized by ST Johnson, an important California based burner manufacturer. The burner reduces NO_x by more than 80 percent. As another example, Altex developed and tested a high efficiency burner for NIECO's meat broilers. This burner has been manufactured and installed in hamburger broilers used in the Burger King and Carl's Junior's fast food chains. Also, in cooperation with Advanced Technology Materials Incorporated, Altex developed the next generation Point of Use (POU) pollution control device for the semiconductor industry. Altex has also teamed with Dewey Electronics, the provider of diesel gen-sets to the military, as a manufacturing partner for its fuel cell power and co-generation systems.

Altex would investigate similar paths for commercialization of BEECH: partnering with a proven manufacturer with existing expertise and manufacturing capabilities, via a joint venture or a licensing agreement. This partnership would then create a product line of systems to meet the demands of a substantial portion of the commercial, large residential and industrial markets. Since the BEECH system integrates with other facility systems (for waste heat, a boiler or process heater; for solar, the solar thermal collector array) of varying type and size, BEECH will not be a one-size-fits-all product, but will be specified on a project-by-project basis, and would likely be incorporated into other building upgrades (for retrofit applications) or as part of the building design/build process. Submittal drawings would be created for each system, to be used for site planning and permitting. Commissioning support staff, either employed by the manufacturer or the installer's trained technicians, would be present at system start-up to train site operators on the new technology.

Dissemination of Results

Before the novel BEECH system can be sold to commercial customers, field demonstration will be required, to evaluate performance and real-world durability. After one or more field tests have been completed, and the feedback used to in system redesigns to improve performance and robustness, a final design and manufacturing plan can be developed. Therefore, the most appropriate near-term method for dissemination of project results will be conference presentations and posters, and articles in peer-reviewed journals. Table 21 presents a list of journals and conferences that have previously supported advanced technologies similar to BEECH (for example, scroll-drive Organic Rankine power systems). If further tests of the BEECH system or related technologies produce more-successful cooling results, this list of potential avenues will be expanded, and submission deadlines and calls for papers will be researched.

Table 21: Potential Publications and Conferences

Type of Publication or Venue	Title	Contact Title	Future Schedule
Trade/Professional Journal	*Journal of Engineering for Gas Turbines and Power*	David Wisler, Editor	Continuous Submittal
Professional Conference	International Refrigeration and Air Conditioning Conference	Kim Stockment, Conference Coordinator	July 2016
Professional Conference	Compressor Engineering, Refrigeration, and Air Conditioning High Performance Buildings	Prof. Eckhard Groll (Chair)	July 2016 (Abstracts due 12/18/15)
Professional Conference	ASME International Mechanical Engineering Congress & Exposition	George Kardomateas (Chair)	November 2016
Peer Reviewed Journal	*Renewable & Sustainable Energy Reviews*	L. Kazmerski, Editor-in-Chief	Continues Submittal
Peer Reviewed Journal	***International Journal of Environmental Engineering***	Prof. Yung-Tse Hung, Editor	Continuous Submittal
Peer Reviewed Journal	***Journal of Energy Resources Technology***	Hameed Metghalchi, Editor	Continuous Submittal

Source: Altex Technologies Corp.

In a longer-term view, BEECH commercialization will require dissemination of results to industry contacts. The Chapter 11 data analysis activities predict the performance of this and other potential models of BEECH. To be commercially successful, cooling capability must be demonstrated, and preferably at a higher cooling output level, to justify the capital cost of a BEECH installation. A preliminary component cost breakdown analysis, based on the Engineering Bill of Materials, has already been prepared and is included in Appendix D. It includes an estimated system cost, and the analyses in Chapter 11 determined system payback time, based on assumed operating costs. All of these data sources would be available to potential manufacturing partners and potential customers (both installers and end users).

Since the purchase and installation of BEECH will represent a capital expenditure for customer sites, it is likely that facility managers will consult with multiple equipment suppliers and installation contractors, so publicizing BEECH to those suppliers and contractors will be critical. Not only will they need to be educated on the unique benefits of BEECH, but also on its use of many familiar, well-known components, such as fan-driven condensers, boiler economizers, and commercially-available solar thermal collector systems, which will decrease resistance to the new technology.

Commercialization and Marketing

The first BEECH system, as built and tested in this project, is of a commercially-viable capacity. It generates a cooling output that can reduce the electrical consumption due to cooling of a small commercial space (for example, the office area of a food-processing company) or fit into the base cooling load of a large residential facility (such as a resort or hotel). In both examples, the valuable hot water also produced by BEECH would be used in the facilities' operations. Since BEECH has been built at this scale, the transition to a commercial product will be easier than from a bench-scale test article.

However, the first BEECH system was necessarily built with additional instrumentation, valves, and fittings, to accommodate the engineering development process. As part of technology transfer, the system must be simplified and re-designed for manufacturing efficiency and low cost. This task is greatly simplified by the large percentage of commercial off-the-shelf (COTS) parts used in its manufacture. Even the scrolls used in the novel expander/compressor can be produced using equipment already in use at scroll compressor manufacturers. Table 22 lists BEECH's main system components, and the change in type or content that is expected as a result of the transition from prototype to production. The potential part count reduction represents an opportunity for lower total component cost, decreased assembly time, and reduced long-term warranty costs. While it is not practical or useful to list every tube, elbow, and fastener in this system at this time, the eventual design for manufacturing activities conducted by Altex's manufacturing or licensing partner would go into that greater level of detail. Appendix A presents a more-detailed BOM of the production system, as a first step in the manufacturing exercise, and includes a part-by-part cost estimate of all key components, based on the purchase prices of the prototype BEECH parts.

Technology Readiness

Many of the BEECH subcomponents have been proven through bench-scale tests, or under operation in the full, assembled prototype system. The expander/compressor has not yet been proven out; as a result, full system operation with both heating and cooling outputs was not demonstrated. Therefore, any promotion of the technology must be preceded by successful performance testing.

Table 22: BEECH Part Content, Experimental and Expected Production System

	Experimental System Content		Anticipated Production System Content	
	Qty.	Type	Qty.	Type
Refrigerant Tank	1	Modified with Sight Glass	1	Designed/sourced with Sight Glass
Refrigerant Flowmeters	2	Piston Style	0	N/A
Refrigerant Filter Dryer and Strainer	2	Replaceable Core Type, Cooper Inline	2	Replaceable Core Type, Cooper Inline
Refrigerant Pump	1	Multi-stage Compound	1	Multi-stage Compound
Variable Frequency Drives	2	AC Drives, 3 Phase Input, Voltage Controlled	2	AC Drives, 3 Phase Input, Voltage Controlled
Expander Compressor	1	Custom Altex Design	1	Commercialized Altex Design
Lubricating Oil Tank	1	Modified with Sight Glass	0	Internal Expander/compressor Oil System
Lubricating Oil Pump	1	Gear Style	0	Internal Expander/compressor Oil System
Lubricating Oil Control Valve	1	Solenoid Type	0	Internal Expander/compressor Oil System
Air Cooled Condenser	1	Finned-tube, Variable Speed Fans	1	Finned-tube, High-efficiency Variable Speed Fans
Brazed-plate-type Heat Exchangers	4	1 Altex HELC 3 Conventional	4	1 Altex HELC 3 Conventional
Controls and User Interface	1	NI LabVIEW (Engineering Interface)	1	PLC with Simplified Touchscreen
Voltage Controlled Refrigerant Valves	4	Sporlan Servo-controlled or Equivalent	3	Sporlan Servo-controlled or Equivalent
Manual/Service Refrigerant Valves	5	Refrigeration Ball Valves	2	Provided for Drier Service
Fill Ports	4	Schrader Type	1	Schrader Type
Refrigerant Sight Glasses	7	Build-in Moisture Indicator	3	Build-in Moisture Indicator
Water Flowmeters	2	Paddlewheel	2	Optional: Paddlewheel; Alternative: Indicator Lights
Water Flow Control and Shutoff Valves	7	Manual Globe and Ball Valve Type	2	Voltage Controlled Valves
Temperature Sensors	26	Thermocouples and thermistors	9	Thermistors

	Experimental System Content		Anticipated Production System Content	
	Qty.	Type	Qty.	Type
Pressure Sensors	9	Refrigeration, water, oil	3	Water and oil pressure sensors replaced with switches/lights for safety/service indication
Solar System Specific		Single Collector (subscale) Single Pump Station Expansion Tank		Thirty Collectors Possible multi-station design Expansion Tank
Waste Heat Specific		Economizer HX Fluid Pump Pre-heater and Tank		Economizer or Other HX sized to Facility Waste Heat Type Fluid Pump Pre-heater and Tank

Source: Altex Technologies Corp.

For BEECH to be sold to commercial customers, field demonstration will be required, to evaluate performance and real-world durability. After one or more field tests have been completed, and the feedback from those tests incorporated, if necessary, into system redesigns to improve performance and robustness, a final design and manufacturing plan can be developed.

The plan can be summarized as follows:

- Complete system testing and demonstrate feasibility of the common-shaft expander/compressor to meet performance targets.
- Create a refined "Alpha" pre-production unit, incorporating lessons-learned from the prototype.
- Complete validation testing of the Alpha in the Altex Test Facility, and then install it at a field demonstration site for performance and durability testing.
- In parallel with Alpha field testing, build at least three expander/compressor units for durability testing independent of the field test.
- In parallel with the latter stages of Alpha-level field testing, design and build at least two production-intent Beta-level systems and perform field testing.
- In parallel with the latter stages of Beta field testing, design and build two pre-production units for completion of Underwriter's Laboratory (UL) and California Office of Statewide Planning and Health (OSHPD) certification tests, using the final design of the expander/compressor
- After successful testing and certification, go to market with commercialization partner.

As noted, Altex would likely commercialize BEECH by partnering with a proven manufacturer with existing expertise and manufacturing capabilities, via a joint venture or a licensing agreement. This partnership would then create a product line of systems to meet the demands of a large portion of the commercial, large residential and industrial markets.

Design for Manufacturing

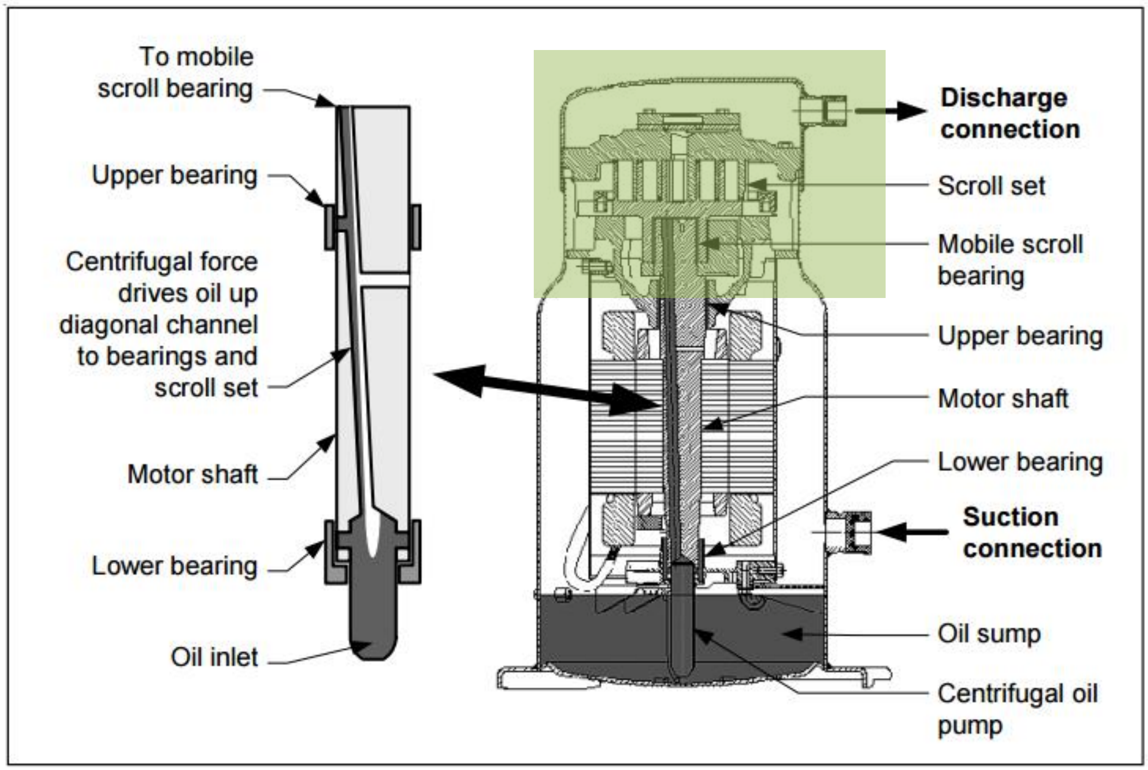
Most components in the BEECH system can be sourced as commercial, off-the-shelf (COTS) parts—already rated and certified for the desired system operating conditions—and so do not require design for manufacturing (DFM) activities. The generator has the potential to be purchased as a COTS brazed-plate heat exchanger (BPHX), or fabricated as a mini-channel heat exchanger using Altex's High Effectiveness Low Cost (HELC) HX technology. HELC is particularly suited for use with advanced, high pressure refrigerants, such as super-critical CO_2. Since initial production versions of BEECH will likely use R-134a or R-1234, HELC is not essential for the Alpha or Beta units described above. The expander/compressor is therefore the primary focus of planned DFM activities.

Scroll compressors are used in mechanical vapor compression (MVC) refrigeration systems, and, as a result, offer industrial reliability. They are also designed to operate with the working fluids and lubricants used in BEECH. The prototype unit built in this project re-used scroll pairs from mass-produced refrigeration compressors as both a compressor and expander, and mated them with a common shaft. The scrolls and shaft are contained within a sealed vessel, and the shaft is supported by a center bearing. A mechanism known as an Oldham coupling, mated to the eccentric feature on the end of the shaft, translates the rotary motion of the shaft into the necessary orbital motion of the scrolls for compression or expansion. Additional journal-type bearings are also located in the shaft-to-scroll interface.

As shown in Figure 70, refrigeration scroll compressors are designed to be mounted vertically, and have a simple oiling system built into the hollow driveshaft. The BEECH expander/compressor is mounted horizontally, with oiled bearings located at both ends of the shaft and in the center. Therefore, the vertical oiling arrangement was not practical. For the prototype, Altex engineers designed a new oiling system with a sump located at the bottom of the vessel, and an external, electrically-driven pump feeding a series of drilled passages in the center housing and the rotating shaft. The oil system also includes a small, air cooled heat exchanger and fan, to control oil temperatures in case unexpected operating conditions were experienced during initial testing.

Unlike the fully-welded and hermetically-sealed production scroll compressor, the prototype expander/compressor was designed with removable vessel heads that are attached using threaded fasteners, and sealed with a reusable O-ring. The system also includes several external solenoid valves and gauges in the oil and refrigerant circuits, to facilitate various tests and anticipated experiments. Refrigerant inlet/outlet connections are Swagelok-brand compression fittings.

Figure 70: Typical Scroll Compressor



Components re-used in BEECH expander/compressor highlighted in green.

Source: Scroll Compressors High Efficiency Compression for Commercial and Industrial Applications, Carrier Corporation, October 2004.

To create a low cost, mass-produced expander/compressor, many of the prototype's serviceable features can be simplified or eliminated, since field dis-assembly and reconfiguration would not be required or desirable. Furthermore, the external, electrically-driven oil pump can be replaced by a pump mechanism internal to the hermetic enclosure, either electrically- or mechanically-driven. Moving the pump inside the vessel eliminates the external plumbing, eliminates the expensive, high-pressure shaft seals, and permits use of refrigerant for oil cooling. The trade-offs between mechanical and electrical pumps, to evaluate cost and reliability, will need to be performed as part of the overall expander/compressor re-design for manufacturing, but the need for an internal pumping mechanism is clear.

The most practical way to manufacture the expander/compressor will be to partner with an existing scroll compressor manufacturer who already has the tooling and machining equipment to create these precision components at low cost and high volume. This leveraged approach would decrease capital investment and take advantage of an existing knowledge base at the manufacturing site. This partner might also supply welding and leak testing services to complete the full expander/compressor assembly, or those final steps could be performed at the same site as the BEECH system assembly. Table 23 summarizes the anticipated differences between the prototype and production expander/compressors, and the required DFM activities.

Required Certification and Testing

The first BEECH system, as built and tested in this project, is of a commercially-viable capacity, and uses many COTS parts already tested and certified for the same purpose as they are being used for in BEECH. For the solar thermal version of BEECH, the solar collectors, pumps and expansion tanks are all commercially available and have existing installation and service infrastructure in place. Similarly, the waste heat version of BEECH will use a code-stamped heat recovery heat exchanger, and a COTS pump and valves. No certification will be required.

The generator, whether purchased as a BPHX or fabricated as a HELC heat exchanger, will carry a pressure vessel code stamp from the fabricator, reflecting a maximum operating pressure in excess of the intended operating pressure. Durability testing of the heat exchanger will be performed as part of the full-system testing.

The expander/compressor will require durability testing as a component, since it is a new design. This will reduce risk to Beta and production testing, and provide additional verification that it will meet customer expectations for service life. Per subcontractor Legacy Chiller Systems, who manufactures refrigeration equipment for the commercial and industrial markets, most customers expect a 20-year service life. To predict the expander/compressor's ability to achieve this goal, accelerated durability testing will be employed. The manufacturing partner is expected to provide substantial inputs on the parameters of those tests, based on their prior experience with scroll devices. At minimum, the test matrix is expected to include repeated start-stops and operation at elevated oil temperatures.

As a whole, the system will require UL certification for electronic components and OSHPD Special Seismic Certification. The former certification is required by most municipal permitting agencies. However, a unique certification for the entire system may not be required if all components are UL-listed, and the controls panel is fabricated and function-tested by a subcontractor certified to UL's industrial control panel (ICP) standards.[20] This will be the intended path for production.

For the Alpha and Beta units, a UL Field Evaluation will be obtained. A UL field evaluation is:

> *"A UL service for evaluating an installed product that has not been previously investigated by UL, or for a UL Listed product that has been modified in the field. Field evaluations are limited to the features and characteristics that can be evaluated at the installed site without damage to the product. […] Product and Site-specific UL Field Evaluations help regulatory authorities determine the compliance of a product, leading to "approval" of the installation. UL's evaluation process consists of documentation review, visual and mechanical inspection, suitability for installation in accordance with the National Electrical Code, applicable testing and an engineering report.*[21]

[20] http://industries.ul.com/blog/become-a-ul-listed-panel-shop.

[21] http://www.ul.com/global/eng/pages/offerings/services/ globalfieldservices/fieldservices/fieldevaluationservices/.

Table 23: Expander/Compressor Design For Manufacturing Activities

Subcomponent	Prototype	Production	DFM Activities	Required Resources and Facilities
Scrolls/Oldham mechanism	Machined with production equipment	No change; optimize expander side for reverse direction operation	Partner with scroll mfg to leverage existing machinery, confirm expander side design	Preferred path is to partner w/ existing mfg to machine scrolls
Center Bearings	Tapered roller, standard size	Journal, standard size	Design bearings and support; perform tribology and durability tests with POE oil	Standard machining and vertical press equipment
Vessel	Removable heads with machined flanges, fasteners, and O-rings	Welded and hermetically sealed vessel	Design new outer shell and welding fixtures	Stamping press, sheet metal roller, robotic welder; leak test station
Connections	Welded Swagelok compression fittings	Brazed tube connections	Sourcing standard parts	Standard braze equipment
Shaft	Lathe machined, milled flats/holes, heat treated, ground	Similar processes; use bar feed lathe and multi-part mill fixtures	Adjust finishes and tolerances for journal bearings	Standard NC machine tools
Structure	Machined billet center section with welded connections and oil sump	Two piece stamping; or one cast & machined/two stamped assembly; welded and post-machined; remove external ports used for lab instruments	Complete re-design for lower cost manufacturing	Outsource stamping and casting; machine w/ standard NC tools; robotic welding w/ dedicated fixtures; precision inspection equipment

Source: Altex Technologies Corp.

After successful completion of the evaluation, a UL Field Evaluation Mark is applied to the product. If the product does not meet the requirements, nonconformance is documented and UL staff can work with the team to bring the product into compliance. Altex has previously obtained these evaluations for custom or modified systems installed in the field, and Altex engineers are familiar with the UL process. Figure 71 shows the typical scope of evaluation, and is consistent with previous, successful evaluations performed by UL for Altex. The evaluation typically covers up to two systems of identical design, and so a total of two evaluations would be obtained: one for the Alpha unit, and one for the two Beta units. As of 2012, cost per evaluation was $4500, and is expected to be similar for BEECH.

Figure 71: UL Field Evaluation Criteria

Summary

Evaluation of equipment will typically include these basic areas:

- Review of equipment drawings
- Review of components for certification and proper application
- Environmental Suitability
- Proper and complete nameplate
- Damaged or worn components
- Grounding and bonding
- Electrical clearances
- Guarding of live parts
- Wiring methods and proper ratings
- Overcurrent protection
- Interlocks and EMO
- Electrical testing may include any or all of the following as applicable for the product and the specific application.
    - Insulation resistance testing of power circuits
    - Heat rise testing at rated load
    - Grounding and bonding continuity testing
    - Dielectric withstand testing where specified by the standard under 'production tests'
    - Leakage current testing on cord and plug connected equipment
    - Interlock functional testing including the EMO

Source: "Equipment Evaluation Overview, Rev. 2." Underwriter's Laboratories Inc. Field Engineering Services. 04/07/2004.

The OSHPD certification is a voluntary certification, but on the advice of Legacy Chiller Systems, should be pursued for BEECH. OSHPD certification is required by many hospitals and institutions that could benefit from BEECH. Special Seismic Certification is a "Certificate of Compliance" provided by manufacturers with assurance that after a Design Earthquake equipment shall maintain structural integrity and functionality.[22] The certification requires, at minimum, shock testing of two units by a testing laboratory that has ISO 17025 accreditation. Alternatively, the testing can be performed at a non-accredited laboratory if it is under the responsible charge of an independent California Licensed engineer. Since OSHPD does not approve test plans, it is usually advisable to hire a professional engineer specializing in these tests to create a test plan, supervise the testing, and prepare the reports for submission. Costs for the certification are expected to be approximately $14,900, broken down as follows:

- Application Review Fee: $5000 (per 2015 OSHPD fee schedule)
- Professional Engineering Services: $8000 (40 hrs @ $200/hr)
- Shake Table Testing: $1900 (Two, one-day tests @ $750/day, plus $200 for certified reports)

Expander/Compressor Design for Manufacturing

As with other aspects of this project, the novel expander/compressor would require the most attention in design for manufacturing. The unit shown in Figure 70 is typical of scroll compressors sold by multiple manufacturers with volumetric flow rates capabilities that match 9,000-600,000 Btu/hr refrigeration systems. Even smaller units are commonly used in passenger vehicle air conditioning systems.

As shown in Table 23, to create a low cost, mass-produced expander/compressor, many of the prototype's serviceable features can be simplified or eliminated, since dis-assembly and reconfiguration would not be required. The most practical way to manufacture the expander/compressor will be to partner with an existing manufacturer who already has the tooling and machining equipment to create these precision components at low cost and high volume. This leveraged approach would decrease capital investment and take advantage of an existing knowledge base at the manufacturing site. This partner might also supply welding and leak testing services to complete the full assembly, or these final steps could be performed at the same site as the BEECH system assembly.

Generator Design for Manufacturing

The BEECH generator transfers heat from the hot heat transfer fluid (for solar, a glycol/water mix; for waste heat, a thermal oil) to the high pressure refrigerant, as shown in Figure 1. For the prototype system, a parallel path was pursued in component selection. To minimize risk and enable earlier assembly of the system, a brazed-plate heat exchanger (BPHX) was specified for the initial system build. Altex engineers used online sizing software from GEA and Alfa Laval to design and select the heat exchanger, and both manufacturers also confirmed the component selection and specifications prior to quotation, particularly the unit's capability to operate at

[22] *OSHPD Special Seismic Certification Preapproval (OSP).* California Office of Statewide Planning and Health. Undated PowerPoint Presentation.

the 445 psia generator inlet design pressure of BEECH. The Alfa Laval unit was chosen, based on lower cost and immediate availability. Upon delivery, the inlet and outlet connections of the heat exchanger were adapted to connect to the high pressure refrigerant tubing. The unit was purchased, at retail pricing, for $2200. Altex's market research indicates that the retail-mark-up on the unit was likely at least 20 percent, and so a wholesale price of $1760 or less is expected for production.

The second of the parallel paths uses Altex's HELC novel mini-channel heat exchanger technology (similar to the unit shown in Figure 28). It is capable of very high internal pressures (in excess of 3500 psia), and as such is suitable for a wide range of working fluids, including super-critical carbon dioxide (ScCO2). BEECH is currently designed to work with R-134a and R-1234, which have global warming potentials (GWP) of 1300 and 4, respectively. While R-1234 is an obvious improvement, it is still a candidate for eventual phase-out. CO_2 has a zero GWP, and is gaining popularity in Europe as a trans-critical refrigerant and as a supercritical working fluid for power cycles. This expanding market can be leveraged to speed production implementation of the HELC technology, and enable BEECH to operate on an even wider range of working fluids.

The robust construction method required for 3500 psia operation is not required for the ~445 psia maximum pressure of BEECH operating near term on R-134a or R-1234. However, the highly effective features of the heat transfer surfaces' geometry are still of benefit to the overall size and weight (and, by extension, material cost) of the generator, and so the brazing trials and test article produced under this project serve a dual purpose of preparing for an eventual future of 0 GWP refrigerants, and performing manufacturing development of low cost heat exchangers that could benefit BEECH in the nearer term.

At this stage of development, it is difficult to compare the cost of a high unit volume, production BPHX generator to a prototype HELC design that provides the same heat duty. Currently, HELC manufacturing uses low quantity manufacturing of plates, frames and inserts as well as a batch process bonding furnace that brazes the HELC units one at a time. In contrast, BPHX use large quantities of stamped sheets with a continuous brazing process that has high throughput and low cost. An improved comparison of HELC and BPHX production costs would require accurate cost estimates for high unit volume production of plates, frames and inserts using dedicated stamping and cutting equipment and large supplies of rolled plates and sheet stock of thicknesses that do not require surface grinding, which was required for tolerance matchup with the prototype parts. Furthermore, to create the inserts with special surface features at low cost, dedicated insert forming machinery is required. In addition to capital costs, the operating costs for this equipment and scrap rates need to be defined, based on the final part geometries. Lastly, unit quantities have to be estimated, considering multiple markets beyond BEECH, to determine the overhead structure costs and quantity material and machining discounts that would be available for a commercial HELC operation. While it is possible to define these costs to facilitate an accurate bottoms-up cost estimate, at this early stage of development it is very difficult to get information from manufacturers to define accurate capital and operating costs for HELC production. In the absence of this type of information, it is useful to define how HELC differs from BPHX designs and fabrication techniques and evaluate the differences in cost as a result of the reduction in material use and fabrication. This difference can be used as an early measure of potential cost savings using the HELC technology.

Relative to fabrication, all of the parts for HELC are braze-bonded in a single operation. This is the same operation as applied to BPHX. As a result of the HELC frame and insert design, the part and joint count for HELC is approximately 75 percent lower than a BPHX of similar heat transfer capacity. Therefore, the quantity of braze material for joining the parts is reduced with HELC. Since the braze compound is more expensive per pound than the base stainless steel material, the reduced need for the compound in HELC will save some cost. However, to be conservative, it is assumed that the HELC and BPHX bonding costs are the same. Also, inlet and outlets bosses and fittings are the same for the two types of heat exchangers.

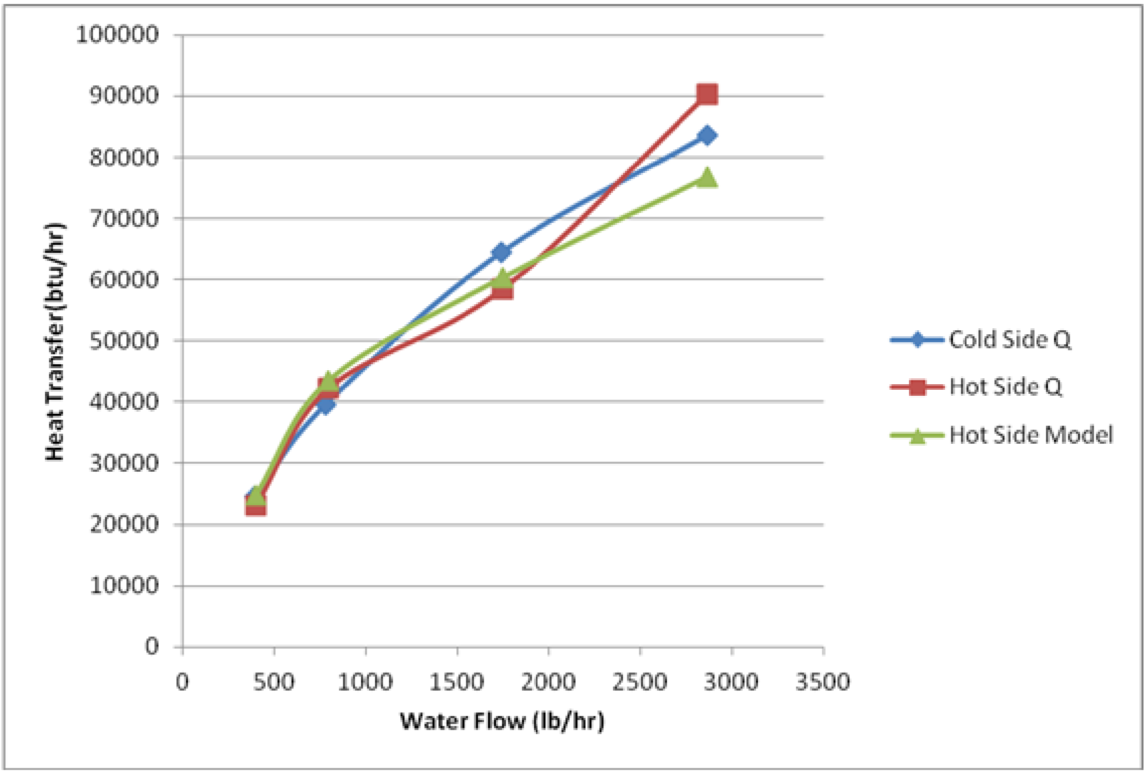
The HELC insert, which is contained within the plates and frames, has a high heat transfer coefficient due to small hydraulic diameter channels and surface features that promote heat transfer. The increased heat transfer performance of the HELC insert versus the plates used in BPHX greatly reduces the core volume, weight and material needed to achieve a given heat transfer. These reductions achieved by HELC can be estimated from the heat transfer model (as presented in the Task 3 report) and test results. Once the material use reduction is calculated, the cost savings for using less material can be calculated. If the material type and bonding materials and processes are the same between HELC and the BPHX, as noted above, then the cost savings for HELC would simply be the material cost savings.

Using manufacturer's data for a BPHX that is compatible with BEECH generator needs, the material use can be derived from the weight data for the given heat transfer performance. For comparison to HELC, the available Altex performance model is used. This model has been validated against data in controlled tests, as shown in Figure 72.

As indicated in the plot, the model and data results are compatible over a range of flow rates. Using this model for the conditions of the BEECH generator, the volume and weight reductions for HELC versus the BPHX can be calculated as a function of the insert height per layer. These results are given in Table 24. The BPHX equivalent insert height is 0.08 inches. Therefore, as the insert height per layer is increased the number of parts and joints for HELC are decreasing. For a 0.08 inches HELC insert height, the parts and joint count would be the same. For the 0.32 inches HELC insert height, the parts and joint count would be reduced by 75 percent relative to a BPHX. Weight and material cost reductions start at 58 percent and decrease to 43 percent.

The current BEECH HELC used a 0.242-inch height insert and has a weight reduction of 50 percent. Using a composite material cost of $3/lb (for stainless steel), the cost reduction for a HELC based generator versus a BPHX of 35 lbs weight would be $53. This cost reduction does not consider any other potential cost reduction (for example, decreased braze alloy) that was noted above. Given the base generator retail price of $2200, this $53 reduction represents a 2.4 percent savings. In a low-margin, competitive marketplace, every small savings can add business or increase margin.

Figure 72: Model Validation: Measured Versus Predicted Heat Transfer



Source: Altex Technologies Corp.

Table 24: Volume and Weight Reductions for HELC

INSERT	VOLUME	WEIGHT
HT/LAYER	REDUCTION	REDUCTION
IN	%	%
0.08	70%	58%
0.16	66%	54%
0.242	60%	50%
0.32	52%	43%

Source: Altex Technologies Corp.

However, the greater value from HELC is in its high pressure capability, which would permit BEECH to use working fluids with zero Global Warming Potential (GWP) and Ozone Depletion Potential (ODP), such as $ScCO_2$. High-pressure testing of the brazed samples will be completed to further validate the HELC potential for those demanding applications.

Standard Components

The BEECH design leverages many commercial off-the-shelf (COTS) parts, particularly from the refrigeration industry. This decreases the need for individual component development and durability testing, and leverages existing high-volume manufacturing processes.

Standard copper tubing and brazing methods were used for all connections except the two high-pressure lines for the pump outlet to generator inlet and generator outlet to expander inlet, which were fabricated from stainless steel tubing with the correct pressure rating. A conventional HVAC installer was contracted to braze the connections, and had no issues. This affirms the assumption that conventional assembly methods can be employed for BEECH production. Minor subcontractor Legacy Chillers, a maker of commercial and industrial chiller equipment, has also reviewed the tubing arrangement and sizing, and has no concerns related to manufacturability.

Reduction of part count is the key DFM activity for the COTS components. Like the expander/compressor, the overall prototype system includes many valves, instruments, and measurement ports not required in a fully-developed commercial system. Some of the valves can be eliminated or changed from electronically-actuated to thermostatic, thus reducing cost and complexity. 12.9 Heat Input: Waste Heat or Solar Thermal

The waste heat variant of BEECH built under this project used a heat recovery heat exchanger (HRHX, also known as an “economizer” in the boiler industry) with a finned coil to transfer heat from the exhaust of natural-gas fired devices, such as boilers and water heaters, to a low vapor pressure thermal oil. The oil can be heated to more than 550 ˚F (288˚C) based on the manufacturer’s ratings. This permits substantial heat recovery from low efficiency equipment (such as legacy boilers), but the BEECH system can still function at maximum designed output with an oil temperature of 400˚F (204˚C), which permits substantial heat recovery from more-modern equipment with lower exhaust temperatures.

The HRHX used in the prototype, shown at left in Figure 73 is essentially an off-the-shelf design from Cain Industries, though the inlet/outlet connections were specified as welded, flanged connections (instead of the standard pipe-thread connections), to provide improved sealing when operated with thermal oil. A similar arrangement would be used in the installation of production versions of BEECH. The exact model of economizer will vary based on the exhaust stack size of the equipment, and the available installation space. As shown in Figure 73, economizers are available in different aspect ratios to fit different installation situations.

A hot oil pump is used to pump the oil through the oil circuit, including the generator. The pump (as well as the variable frequency drive that controls its speed) used in the prototype is a COTS part and could be used again in production installations. The remainder of system components include shut-off valves, a heated expansion tank, and temperature sensors. These would all be installed as-appropriate in the facility upgrade for BEECH, and their cost will depend somewhat on the physical location of the main BEECH system in relation to the waste heat source.

Figure 73: Heat Recovery Heat Exchangers Installed at Altex (left) and Multi-unit Field Installation (right)

Source: Altex Technologies Corp

Source: Cain Industries

The solar thermal variant of BEECH will use solar thermal collectors and a working fluid of propylene glycol/water mixture. All of the components used in the BEECH subcomponent tests were COTS parts, and experienced no reliability issues during testing. In fact, the installation followed all manufacturer recommendations for typical solar thermal installations, even though the working temperature of the glycol mix was hotter than is typically needed for a hot water system. The only added component therefore was a buffer tank above the expansion tank, to ensure durability of the expansion tank diaphragm at the high collector outlet temperatures. The subcomponent test was performed with a Kingspan Thermomax collector, though other manufacturer's evacuated-tube collectors could be substituted. The pricing breakdown in this report assumes Kingspan components. Each Thermomax collector is a 30-tube array and can generate a peak output of 10,000 Btu/hr.

Therefore, at least 25 panels would be required to meet the thermal input needs of a system with the same thermal output as the prototype BEECH (5.0 tons cooling/44 kWt heating). Allowing some margin for production of maximum output at non-peak insolation times, Altex engineers specified a 30 collector array. Production costing activities proceeded with the same assumption.

System Assembly and Testing Process

To evaluate the activities and costs of system assembly and test, Altex consulted with subcontractor Legacy Chiller Systems' President Martin King. He reviewed the assembled prototype system in person, as well as the Piping and Instrumentation Diagram. Based on these inputs, and his knowledge of typical manufacturing and testing of refrigeration equipment, Mr. King created the assembly plan and hours estimate presented in Table 25.

Table 25: BEECH Assembly Process and Time

Action	Labor Hours
1. Preload	
Assemble frame and component support structures. Pull all components from inventory into loading area.	4.5
2. Load	
Set all major components in place. Each item will be orientated for plumbing connections.	6
3. Piping	
Pre-bend all tubing w/programmed bender. Each plumbing section will be placed and connected to components. Valves, driers etc, will be placed into position by hand. Some plumbing supports will also be set at this time.	9
4. Brazing	
Braze all plumbing connections using 15% Sil-phos. Flow dry nitrogen while brazing.	5
5. Electrical	
Mount pre-built panel(s) and wire to electrical components.	9
6. Leak test and charge	
All systems tested with dry nitrogen. Leaks will be found and fixed.	3
7. Charging	
All systems will be charged by weight by a licensed technician.	3.5
8. Testing	
Procedure to be developed based on final system configuration. Estimate based on a chiller with about the same number of components.	5
9. Ship prep	
Palletize, wrap and build hard wood (slat) enclosure for machine.	5
Total	50

Source: Altex Technologies Corp

The total estimated assembly and test time is 50 hours. Considering that some work can be done by less-skilled technicians, and that others must be performed by certified Refrigeration Technicians, a loaded technician labor rate of $42/hour is assumed, yielding a total assembly cost of $2,100.

Estimated System Cost

For estimation of system cost, the BEECH system is considered as a base system, plus either a waste heat or solar thermal input subsystem. The base would be similar to the system previously shown and, for a given system output, would be independent of heat source. The waste heat or solar thermal subsystems are priced separately as "typical installations", though installation content will vary with installation site. Appendix D contains the results of the analysis.

Implementation and Investment

At this stage of development, the BEECH system performance and durability has not been proven. For BEECH to be sold to commercial customers, field demonstration will be required, to evaluate performance and real-world durability. After one or more field tests have been completed, and the feedback from those tests has been incorporated, if necessary, into system redesigns to improve performance and robustness, a final design and detailed manufacturing plan will be developed. The plan is detailed in the earlier section on technology readiness.

Once a partnership with a proven manufacturer with existing expertise and manufacturing capabilities was formed, via a joint venture or a licensing agreement, Altex and the partner would work together to define a more accurate and more detailed plan, particularly in regards to the expander/compressor. As shown in Appendix D, heat exchangers make up a substantial portion of the system price, and Altex's heat exchanger expertise could be leveraged to design improved heat exchangers that would be manufactured under the partnership at lower cost. The optimum partner will have existing scroll manufacturing and refrigeration system assembly/test capabilities, thus reducing the capital costs associated with starting production. Not including the durability and demonstration testing activities noted above, which are considered developmental testing, a capital cost of less than $500,000 is feasible, since the only tooling required would be for the expander/compressor components and assembly processes that are not common to a scroll compressor.

Hazardous or Nonrecyclable Materials

The system, as designed, contains no hazardous or non-recyclable materials. No fasteners require cadmium or hexavalent chrome coatings. Refrigerant use and service is regulated by the US Environmental Protection Agency (EPA), and all service of the refrigeration components should be performed by a technician certified as Type I Technician under EPA Clean Air Section 608. The solar thermal variant of BEECH uses a non-toxic glycol/water mix as the heat transfer fluid. The waste heat variant uses a synthetic mineral oil. Both are common working fluids with standard Material Safety Data Sheets. Both have established processes in place in industry for disposal and re-use.

Technology Readiness and Commercialization Conclusions

At the time of this report, the novel BEECH system is still in the engineering development stage, and will require Alpha and Beta units to validate the novel system, and provide valuable field test results. The use of COTS components, even in the prototype, reduces the need for substantial DFM activities. The expander/compressor requires the most DFM focus, and those activities would be pursued with the cooperation of the manufacturing partner. Certification by UL and OSHPD are also required, to maximize the market for BEECH, and to assure that regulatory and municipal permitting agencies will allow BEECH to be installed.

Conclusions

The BEECH project pursued a novel combination of an organic Rankine power cycle and refrigeration cycle. Analysis has shown this to be a practical development which can be operated on at least one readily-available working fluid (R-134a). The projected output of this system is 5.0 tons of refrigeration and 190,000 Btu/hr of water heating, both of which can offset the energy needs of commercial, industrial, and large residential buildings. This system could also be scaled up or down to meet other heat source magnitudes. Based on projected capital and operating costs, the waste heat-driven system of the noted capacity will have a four-year payback time. The solar-driven version will have a thirteen-year payback time, with current incentives applied.

The Site Specification work showed that hot water demand is the limiting capacity factor. A site with a continuous supply of waste heat of greater than 500,000 Btu/hr (or a large area for solar thermal collector installation) may not be able to use the amount of hot water that could be generated by a larger BEECH system. The ratio of hot water energy to cooling energy produced by the BEECH cycle is >3:1, but this ratio is not necessarily matched by our analysis of typical building demands. For sites with higher hot water demand, a larger-capacity BEECH system would be appropriate, or the system could be re-configured to use water to partially or fully replace the duty of the air cooled condenser in the power cycle.

The technical challenges of the common-shaft expander/compressor were considerable, and the inconsistent starting behavior prevented testing of the full system at steady state. Previous university and research projects have postulated that this device could be built from existing scroll compressors, or have built only the expander section, and then postulated how the compressor could be mated to it. The BEECH project has advanced the state of the research to a full-scale, integrated unit, which has shown promising initial results, but which was not operated continuously. The low cost of scroll devices, as compared to high-speed turbo machinery, still make them attractive as the basis for this cycle, as well as conventional Organic Rankine power cycles, but further development time and funding would be required to field a fully functional prototype system.

LIST OF ACRONYMS

Term	Definition
ASHRAE	American Society of Heating Refrigeration and Air Conditioning Engineers
BEECH	Building Energy Efficient Cooling and Heating
BTU	British thermal units
BOM	Bill of materials
BPHX	Brazed plate heat exchanger
CAD	Computer-aided design
CEUS	Commercial End Use Survey
CHEMCAD	A commercial software package used in chemical process modeling
CO	Carbon monoxide
CO2	Carbon dioxide
COTS	Commercial off-the-shelf
CPR	Critical project review
CPUC	California Public Utilities Commission
DFM	Design for manufacturing
DOE	United States Department of Energy
EPA	United States Environmental Protection Agency
EPDM	Ethylene propylene diene monomer, a roofing membrane material
GPM	Gallons per minute
GWP	Global warming potential
HELC	High-effectiveness, low-cost heat exchanger
HRHX	Heat recovery heat exchanger
HVAC	Heating, ventilation, and air conditioning
HX	Heat exchanger
ICP	Industrial Control Panel, a standard of UL
IOU	Investor-owned utility
MMBtu/hr	Millions of Btu's per hour; 1 MMBtu = 10 therms
MVC	Mechanical vapor compression
NOx	Oxides of nitrogen
NREL	National Renewable Energy Laboratory
ODP	Ozone depletion potential
OSHPD	California Office of Statewide Planning and Health

Term	Definition
PG&E	Pacific Gas and Electric Company
POC	Products of combustion
PSI	Pounds per square inch
PSIA	Pounds per square inch, absolute
RFQ	Request for Qualifications
SCAQMD	South Coast Air Quality Management District
SCE	Southern California Edison Company
UL	Underwriter's Laboratory
VPE	Vacuum Process Engineering, a Sacramento manufacturer.
VFD	Variable frequency drive

APPENDIX A: Summary of System Validation Tests

To verify the system's readiness for testing, Altex engineers and technicians followed the System Test Plan, proceeding subsystem-by-subsystem, to verify leak tightness and ability to meet the system design operating points (see Table A-1). The latter was usually satisfied by a successful flow test of the individual subsystem. As expected, minor leaks were found, but all could be resolved.

Chilled Water/Hot Water Loop

Altex engineers conducted a series of tests confirming that the chilled water and hot water circuits would perform as required. The circuits were pressurized to 25 psig with compressed air. A soapy water solution was applied at all joints and then visually inspected. Leaks were identified at the water flow meter connections and pipe unions. After fixing the leaks, the system was left pressurized for one hour to determine pressure decay with time. A few threaded connections showed signs of slow leaks, and so were resealed and checked, allowing the system to pass the pressure decay re-test.

To verify flow capability, the fifteen-ton chiller was used to flow water. No leaks were seen, and the max system targets of 3.7 gpm and 12.2 gpm were achieved for the hot water and chilled water, respectively, by adjusting the globe valves on the system and the bypass valve on the chiller. The water flow meter indicators were calibrated to show a red status light when the flow is 20 percent below the set-point.

Oil Loop (in system)

The oil loop piping within the BEECH system was first leak tested with air at 25 psig. Soapy water solution was applied to visually inspect for leaks at joints. A few leaks were found at pipe unions, particularly the connections to HX-3, but were easily remedied. A flow test with Therminol 55 heat transfer oil was performed after the system and facility oil loops were connected, and no leaks were found.

Refrigerant Pump

The refrigerant pump was installed on the Altex refrigerant pump test bench with the Max Machinery piston flow meter, Sporlan pressure transducers, and other necessary data acquisition. Data was collected at 30Hz, 45Hz, and 60Hz. As noted above, pump curves were constructed for 30Hz and 45Hz. At 60Hz the pump was able to produce 308 psid at 4.3 gpm satisfying the requirement of 300 psid at 3.16 gpm.

Refrigerant System

The refrigerant pump was installed in the BEECH refrigerant system, and the system was initially tested with compressed nitrogen at 150 psig. Soapy water was applied to visually inspect for leaks. After the soapy water leak test was passed, engineers pressurized the system to 150 psig, using one pound of R-134a refrigerant and the balance nitrogen. Tiny leaks could then be found with an electronic detector specifically designed for refrigerants. Leaks were fixed at O-ring interfaces. The decisive test for a refrigeration system is a vacuum test. A vacuum level of fewer than 500 microns indicates a suitable degree of leak tightness. The system was evacuated using standard HVAC equipment, and achieved 470 microns vacuum.

Oil Loop (facility-side heat recovery)

The portion of the oil loop associated with the boiler heat recovery was leak checked with 25 psig air and soapy water solution. The resulting leaks were fixed and the system passed a 25 psig pressure-decay test. Engineers then added oil and pumped flow only through the heat recovery portion. No leaks were found. After the BEECH system was connected, oil was circulated through the entire oil circuit. Visual inspection found no leaks. The oil pump achieved the required capacity of 8.7 gpm at a speed of 41Hz. It was found that pumping Therminol 55 at room temperature could draw excessive current and fault the VFD. The system includes an oil preheater to raise the temperature of the oil to reduce its viscosity at start-up. The oil preheater function was tested by running the oil pump at 15Hz, and setting the heater controller to 100°F. It took approximately 20 minutes for the temperature in the oil tank to reach 100°F, and after this time, the pump current draw was not a problem. This procedure will be followed during subsequent system tests. It is important to note that this electrical heater operation will only be required during cold starts, and will not affect steady-state system efficiency.

Expander/Compressor

After final assembly, the expander/compressor was pressurized with one pound of refrigerant and the balance nitrogen. The electronic detector found no leaks. Initially, the vacuum pump was not able to vacuum down the assembly to under 500 microns. This was attributed to degraded vacuum pump oil. After servicing the pump, a vacuum level of fewer than 500 microns was achieved.

Heat Recovery Heat Exchanger

Economizer operation was confirmed by heating the unit's control thermocouple with a heat gun and visually confirming the response of the damper. Engineers found that the controller was missing a 500-ohm resistor to convert the 4 to 20mA signal into 2 to 10VDC accepted by the damper. In addition, the manufacturer's drawing specified 24VAC power for the controller, but supplied a 120VAC model. Once the electrical issues were resolved, the damper behaved as expected with full range of motion.

Table A-1: System Test Results

Component	Parameter	Measurement Method	Test Criteria	Pass/Fail
Water Loop	Pressure Test (Air)	Visual Inspection	No soap bubbles @ 25 psig min	Pass on retest 8/11/15
	Pressure Test (Air)	Pressure decay @ 30 psig	<0.5 psi drop in 1 hr @ 25 psig min	Pass on retest 8/12/15
	Flow Test—leak tightness	Visual Inspection	No visible leaks	Pass 9/11/15
	Flow Test—flow rate of heated water	Proteus paddlewheel flowmeter	Min. 3.7 gpm flow	Pass 9/11/15
	Flow Test—flow rate of chilled water	Proteus paddlewheel flowmeter	Min. 12.2 gpm flow	Pass 9/11/15
Oil Loop (in system)	Pressure Test (Air)	Visual Inspection	No soap bubbles @ 25 psig min	Pass on retest 8/14/15
	Flow Test	Visual Inspection	No visible leaks	Pass 9/18/15
Refrigerant System	Pressure Test (N2)	Visual Inspection	No soap bubbles @ 150 psig min	Pass on retest 9/15/15
	Pressure Test (R-134a)	Electronic Leak Detector	< 3 bars on meter (@ 150 psig pressure)	Pass on retest 9/16/15
	Vacuum Test	Vacuum Gauge	< 500 microns	Pass 9/18/15
Refrigerant Pump	Flow (bench test)	Sporlan Pressure Sensor MAX piston flowmeter	Min. 3.16 GPM @ 300 psid	Pass 8/19/15
Oil Loop (heat recovery)	Pressure Test (Air)	Pressure Decay @ 30 psig	<0.5 psi drop in 1 hr @ 25 psig min	Pass on retest 8/12/15
	Flow Test—flow rate	Manual Flow Meter	Min 8.7 gpm	Pass 9/18/15
	Oil preheat temp	J-type thermocouple	Min 100 F	Pass 9/15/15

Component	Parameter	Measurement Method	Test Criteria	Pass/Fail
Expander/ Compressor	Pressure Test (R-134a)	Electronic Leak Detector	< 3 bars on meter (@ 150 psig pressure)	Pass 9/22/15
	Vacuum Test	Vacuum Gauge	< 500 microns	Pass on retest 9/24/15
Economizer	Damper Operation	Visual Inspection	Full range of motion*	Pass on retest 9/23/15

***Economizer outlet temperature variation simulated using a heat gun, measured with the system's J-type thermocouple**

Source: Altex Technologies Corp

APPENDIX B:
Hot Water Production Sample Calculation Worksheet

TEST CONDITIONS

Temp of POC into Econ	499	F		
Temp of POC out Econ	390	F		
Furnace out temp	1317	F	1777	R

BOILER FLOWS

Fuel flow rate	2301	scfh	
%O2	8	%	
Theoretical air	1.55		
Air flow	34247	scfh	
Ambient Temp	70	°F	
Density of air	0.075	lb/ft^3	
Mass flow of air	2569	lb/hr	
Mass flow fuel	104	lb/hr	
Burner total mass flow	2672	lb/hr	
Dilution Air Cp	0.240	Btu/lb°F	
Furnace temp out	1317	°F	Measured
Flue gas Cp	0.299	Btu/lb°F	POC weighted
Economizer temp in	499	°F	
Economizer temp out	390	°F	
Dilution blower flow	6342	lb/hr	
Total mass flow	9014	lb/hr	

POC Cp

Substance	Mass Flow (lb/hr)	Cp (Btu/lbmol)	Molar Mass (lbm/lbmol)	Cp (Btu/lb)
POC	2755	---	---	0.299
Carbon Dioxide (CO_2)	278	12.98	44	0.295
Water (H_2O)	221	9.78	18	0.543
Nitrogen (N_2)	2038	7.75	28	0.277
Oxygen (O_2)	218	8.30	32	0.260
Non water mass	2534	---	---	0.277
POC + Dilution air				0.260

ENERGY CALCULATIONS

Energy in exhaust calc--actual test conditions		
Burner mass flow	2672	lb/hr
Water mass flow	221	lb/hr
Remainder ("air")	8793	lb/hr
Econ in temp	499	F
Ambient temp	70	F
Econo out temp	390	F
Heat recovery	255,502	Btu/hr

SAVINGS

3.7 gpm data						
ECONOMIZER						
Oil	62.6	kW	213,654	Btu/hr		
Difference	12.3	kW	41,849	Btu/hr	16.4	%
HX-3						
Oil	14.4	kW	49,147	Btu/hr		
Water	14	kW	47,782	Btu/hr		
Difference	0.4	kW	1,365	Btu/hr	2.8	%
Natural gas saved	14.3	therms/day				
NOx	9	ppm				
k	1.194E-07					
Fd	8710					
O2	3	%				
NOx	0.011	lb/MMBtu				
NOx avoided	0.004	lb/day				
CO	50	ppm				
k	7.264E-08					
Fd	8710					
O2	3	%				
CO	0.037	lb/MMBtu				
CO avoided	0.013	lbs/day				

APPENDIX C: BEECH Payback Calculation References

To determine a 2015 natural gas price, Altex and subcontractor Oxford Engineering researched the published rates of the three Investor Owned Utilities (IOU), and determined an aggregate gas price based on the predicted consumption of the BEECH system or its equivalent. The results are summarized in Table C-1:

Table C-1. 2015 Investor-Owned Utility Natural Gas Prices

PG&E Natural Gas Rate ($/therm)					
Summer		Winter		Average Rate	
First 4000 therm	Excess	First 4000 therm	Excess	First 4000 therm	Excess
$0.81656	$0.55584	$0.92428	$0.59924	**$0.87042**	$0.57754

So-Cal Gas Natural Gas Rate ($/therm)			
Tier I (first 250 therm)	Tier II (up to 4167 therm)	Tier III (above 4167 therm)	Average Rate for first 4000 therm
$0.87049	$0.61463	$0.44308	**$0.63062**

SDG&E Natural Gas Rate ($/therm)		
0 to 1000 therm	1001 to 21000 therm	Average Rate for first 4000 therm
$0.74796	$0.58079	**$0.62258**

Natural Gas Rate ($/therm)	
CEC PON-12-503	**Average 2015 Actual**
$0.68	**$0.71**

To determine a 2015 electricity price, Altex and subcontractor Oxford Engineering researched the published rates of two IOU's, and determined an aggregate price based on the predicted consumption of the BEECH system or its equivalent. The results are summarized in Table C-2:

Table C-2: 2015 Investor-Owned Utility Electricity Prices

PG&E Electricity Rate ($/kWh)							
Non-Time-of-Use Rate		Time-of-Use Rate					Average Rate
Summer	Winter	Peak Summer	Part-Peak Summer	Off-Peak Summer	Part-Peak Winter	Off-Peak Winter	
$0.2398	$0.1625	$0.2604	$0.2511	$0.2227	$0.1728	$0.1530	**$0.2089**

SDG&E Electricity Rate ($/kWh)				
Summer		Winter		Average Rate
Primary	Secondary	Primary	Secondary	
$0.12847	$0.12888	$0.12847	$0.12888	**$0.12868**

Electricity Rate ($/kWh)	
CEC PON-12-503	**Average 2015 Actural**
$0.13	**$0.17**

The sources of this data are as follows:

- Solicitation Values, natural gas and electric, CEC PON-12-503, Appendix 17:
    - Solicitation Utility Cost: Natural Gas Rate: $0.68/therm

    - Electricity Rate: $0.13/kWh

- 2015 Actual Utility Costs for Commercial
    - Pacific Gas & Electric
        - Natural Gas Rate: $0.87/therm[23]
        - Electricity Rate: $0.21/kWh[24]
    - Southern California Gas Company
        - Natural Gas Rate: $0.63/therm[25]
    - San Diego Gas & Electric
        - Natural Gas Rate: $0.62/therm[26]
        - Electricity Rate: $0.17/kWh[27]

The incentives for solar thermal equipment are based on the CPUC California Solar Initiative - Thermal Program.[28] Incentives were calculated using the CSI—Thermal Program Incentive Calculator,[29] as shown in Figure C-1. The estimated annual energy savings is 3303 therms, resulting in an estimated incentive of $66,688. It should be noted that incentive rates will decline over the life of the program in four steps to facilitate market transformation. As of December 19, 2015, Pacific Gas and Electric (PG&E), SoCal Edison (SCE), and SoCalGas all have remaining funding of more than $66,688. Therefore, we are still in Step 1 incentive rate, which is $20.19/therm. However, the incentives for electricity savings have been exhausted for commercial systems. Therefore, only the natural gas incentive is applicable to the BEECH solar system.[30]

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[23] PG&E Gas Rate (Sheet 1): http://www.pge.com/tariffs/tm2/pdf/GAS_SCHEDS_G-NR1.pdf.

[24] PG&E Electricity Rate (Sheet 3): http://www.pge.com/tariffs/tm2/pdf/ELEC_SCHEDS_A-1.pdf.

[25] SCG Gas Rate (Sheet 2): https://www.socalgas.com/regulatory/tariffs/tm2/pdf/G-10.pdf.

[26] SDG&E Gas Rate (Sheet 2): http://regarchive.sdge.com/tm2/pdf/GAS_GAS-SCHEDS_GN-3.pdf.

[27] SDG&E Electricity Rate (Sheet 1): http://regarchive.sdge.com/tm2/pdf/ELEC_ELEC-SCHEDS_A.pdf.

[28] CSI – Thermal Program: http://www.cpuc.ca.gov/PUC/energy/Solar/swh/csithermalincentives.htm.

[29] CSI – Thermal Program Incentive Calculator: https://www.csithermal.com/calculator/commercial/result/bf79481e-390f-4d25-956b-3a4235cbf652/.

[30] CSI Solar Thermal Program Incentive Step Tracker: https://www.csithermal.com/tracker/.

Figure C-1: California Solar Initiative Thermal Program Incentive Calculator Output

GO solar CALIFORNIA (http://www.gosolarcalifornia.ca.gov/)

California Solar Initiative T

Home (/) | Register (/registration/) | Calcul

| Eligible Contractors (/eligible_contractors/)

Incentive Calculator

Thank you. The results for project "Altex Tech - Ken" are ready. Please see the table below for the estimated incentive f
program step. The current program steps are also displayed below.

Annual Energy Savings: 3303 Therms

Installed Capacity: 89.58 kW_{therm}

Step	Incentive Rate	Estimated Incentive
1	$20.19	$66,688.00
2	$17.16	$56,679.00
3	$10.15	$33,525.00
4	$3.13	$10,338.00

Program Administrator	Current Step
PG&E	1
SoCalGas	1
CSE	1

Collector Effective Date: Feb. 7, 2015

Simulation Status: Your simulation is complete.

Simulation Finish Time: Dec. 21, 2015, 2:43 p.m.

Current Simulation Engine Version: v11 - December 12, 2014 (/trnsys_release_notes/#v11 - December 12, 2014)

Engine Version Used for This Simulation: v11 - December 12, 2014 (/trnsys_release_notes/#v11 - December 12, 20

Applying for an incentive for this system?

If you plan to apply for an incentive for this system, simply log in and return to this page (/calculator/commercial/result/b
3a4235cbf652/).

Project Details

Heat Exchanger(s): External Supply Side w/ External Load Side

Freeze Protection: Glycol

Overheat Protection: Advanced Controller with a Thermal Cycling Function

Tank Configuration: Solar and Auxiliary Storage are the same Tank

Collector: Kingspan Environmental Ltd. - DF100-30 (SRCC 2009049C)

California Solar Initiative - Thermal Program \PZZZZZZ F https://ww

Number of Collectors: 30

Average Collector Module Area: 45.94

Number of Collectors in Series per Flow Path: 4

Total Solar Storage Capacity: 17.04

Total Number of Solar Tanks: 3

Total Backup Storage Capacity: 3.9

Total Number of Backup Tanks: 3

Backup Fuel Source: Gas

Maximum Auxiliary Heat Capacity: 194000

CEC Climate Zone: Z04

Building Type: Hotels/Motels

Other Building Type Description:

Load Profile: Hotels/Motels

Load Profile Description File:

Hot Water Demands (gal./day): 5130

Measurement for GPD benchmark: 342

Recirculation Loop: True

Set Point Temperature for Back-up Water Heater: 140°

Set Point Temperature for delivered water: 140°

Tracking: Fixed Surface

Array Tilt: 30°

Array Azimuth: 137°

Average Annual Access: 100%

Project Name: Altex Tech - Ken

For assistance, email support@csithermal.com (mailto:support@csithermal.com) or call 888-323-3133 during normal

APPENDIX D:
Anticipated BEECH Production Bill of Materials

The following tables present a preliminary Bill of Materials for a 5 tons cooling/44 kW water heating system. Costs are based on prices Altex paid for the parts during this project, without any wholesale discount. As such, they include a similar retail mark-up as Altex would include for the system when sold as a unit.

Table D-1: BEECH Base Assembly Price Roll-up

Qty	Unit	Component	Manufacturer or Vendor Price Source	Production Price
		City Water Supply		**$ 1,669**
1	EA	Brass Solenoid Valve	McMaster	$ 175
1	EA	Brazed plate HX	Alfa Laval/Legacy	$ 769
1	EA	Brazed plate HX	Alfa Laval/Legacy	$ 655
1	EA	Temp sensor (purchased TC price used as ref.)	McMaster	$ 35
1	EA	Water hammer arrestor	Grainger	$ 35
		Chilled Water Production		**$ 1,332**
1	EA	Brass Solenoid Valve	McMaster	$ 175
1	EA	Temp sensor (purchased TC price used as ref.)	McMaster	$ 35
1	EA	Brazed plate HX	Alfa Laval/Legacy	$ 1,122
		R-134 (Power Loop)		**$ 31,232**
1	EA	Refrigerant Tank	Henry/RSD	$ 395
1	EA	Filter dryer (Sporlan Catch-All)	Sporlan/RSD	$ 204
1	EA	Core part (Sporlan Catch-All)	Sporlan/RSD	$ 42
1	EA	Filter element (Sporlan Catch-All)	Sporlan/RSD	$ 28
2	EA	Mounting bracket (Sporlan Catch-All)	Sporlan/RSD	$ 42
2	EA	Ball valve, 5/8" (before & after filter dryer)	RSD	$ 54
1	EA	Sight glass, 1/2 ODF (Sporlan See-All)	Sporlan/RSD	$ 23
1	EA	Refrigerant pump	Speck Pump	$ 8,950
1	EA	VFD refrigerant pump	Lenze AC Tech	$ 488
1	EA	Refrigerant flow control valve	Sporlan/RSD	$ 510

Qty	Unit	Component	Manufacturer or Vendor Price Source	Production Price
1	EA	Interface board for EEV, IB2Q	Sporlan/RSD	$ 215
1	EA	Temp sensor (purchased TC price used as ref.)	OMEGA	$ 32
1	EA	Pressure sensor, R134a, 0-500 psi	Sporlan/RSD	$ 135
1	EA	1/4 SAE 45 flare x 1/4 hose barb, stainless steel	Fast Fittings	$ 4
1	EA	Expander/Compressor	Altex/Copland	$ 2,500
1	EA	HX-1 Condenser, fans and controls	BOHN/ Legacy	$ 16,563
1	EA	Temperature sensor, R134a, 3k brass	Sporlan/RSD	$ 75
1	EA	Bypass control valve	Sporlan/RSD	$ 475
1	EA	Control valve cable, 20' length	Sporlan/RSD	$ 70
1	EA	Interface board for EEV, IB2Q	Sporlan/RSD	$ 215
1	EA	Sight glass, 3/8 ODF (Sporlan See-All)	Sporlan/RSD	$ 19
4	EA	Vibration Dampening mounts	McMaster	$ 58
1	EA	Pressure sensor, R134a, 0-300 psi	Sporlan/RSD	$ 135
		R-134 (Refrigeration Loop)		**$ 1,879**
1	EA	Filter/Strainer, 3/8 in and out	RSD	$ 5
1	EA	Sight glass, 3/8 ODF (Sporlan See-All)	Sporlan/RSD	$ 19
1	EA	Expansion Valve	Sporlan/RSD	$ 255
1	EA	Interface board for EEV, IB2Q	Sporlan/RSD	$ 215
3	EA	Temperature sensor, R134a, 3k brass	Sporlan/RSD	$ 225
1	EA	Pressure sensor, R134a, 0-300 psi	Sporlan/RSD	$ 135
1	EA	Pressure sensor cable	Sporlan/RSD	$ 50
2	EA	Fill valve, R134a, 3/8" tube	JB/RSD	$ 20
1	EA	Oil Separator	Henry/RSD	$ 154
1	EA	PLC with Touch screen (IDEC or similar)	Legacy	$ 500
1	EA	Electronics/wiring		$ 300
		Tubing, fittings, and solder/braze		**$ 600**
		Total Component Cost		**$ 36,711**
		Assembly Labor		**$ 2,100**
		Estimated BEECH Base System Price		**$ 38,811**

Table D-2: Typical BEECH Waste Heat Recovery System

Qty	Unit	Component	Manufacturer or Vendor Price Source	Production Price
1	EA	Manual shut off, 1" NPT ball valve	McMaster	$ 22
1	EA	Oil Tank--7 gal	McMaster	$ 294
1	EA	Immersion heater--3 kW	McMaster	$ 130
1	EA	Oil-level indicator with shutoff valve	McMaster	$ 103
2	EA	Temp sensor (purchased TC price used as ref.)	McMaster	$ 70
1	EA	Wye strainer	McMaster	$ 25
1	EA	Pump	Viking	$ 3,279
1	EA	Speed Controller	AC Tech	$ 250
1	EA	Manual control valve, 1" NPT ball valve	McMaster	$ 22
1	EA	Economizer, w/ controller	Cain Industries	$ 15,218
2	EA	Manual drain, 1" NPT ball valve	McMaster	$ 44
1	EA	Pressure relief valve	McMaster	$ 119
1	EA	Brazed plate HX	Alfa Laval/Legacy	$ 2,237
1	EA	1/16 DIN temp controller	McMaster	$ 226
		Heat Recovery Price, w/ typical content		**$ 22,040**

Table D-3: Typical BEECH Solar Thermal Collection Subsystem Price Roll-up

Qty	Unit	Component	Manufacturer or Vendor Price Source	Unit Price	Total Price
30	EA	DF-100 30 tube manifold	Kingspan	$ 967	$ 29,022
90	Case	DF-100 evacuated tubes, per 10	Kingspan	$ 1,144	$ 102,998.70
30	EA	Flat roof fixing Kit (A-Frame)	Kingspan	$ 639	$ 19,182
30	EA	Connection kit	Kingspan	$ 136	$ 4,068
2	EA	S16 pump station	Kingspan	$ 755	$ 1,509
3	EA	Zilmet expansion tank (6.6 Gal)	Kingspan	$ 216	$ 648
3	EA	Zilmet intermediate tank (1.3 Gal)	Kingspan	$ 130	$ 391
Solar Thermal subsystem Price w/ typical content					**$ 157,819**